SOME MODERN METHODS OF VENTILATION

WITH SPECIAL REFERENCE TO PUBLIC BUILDINGS

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STANDARDS OF VENTILATION, DESIGN OF DUCTS, SELECTION OF FANS, WASHERS, AND HEATERS, SPECIFICATIONS, TEST FORMS, AND SPECIMEN SCHEDULES FOR DESIGNERS

BY

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WITH ILLUSTRATIONS, CHARTS, AND TABLES

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FOREWORD

The object of the present work is to present, in as simple and concise a form as possible, the general principles and practice of design of modern ventilating plant.

No attempt has been made to deal exhaustively with the theoretical principles involved in the treatment of air, the design of fans, air washers, &c., but it is hoped that the information given will enable health authorities, architects, heating, sanitary and electrical engineers, metal plate workers, and others called upon from time to time to deal with ventilating apparatus from the fixing of a twelve-inch fan to the design or specification of a small but complete installation, intelligently to apply, and with greater efficiency than hitherto, the apparatus placed at their disposal by the various manufacturers.

The subject is a complex one, and because of its many intricacies cannot be fully dealt with here; but readers may be inclined to pursue their investigations in the more extensive works published on the subject, notably "Barker on Heating and Ventilating" (Carton Press), "Mechanical Equipment of Federal Buildings," by N. S. Thompson (David William's Co., U.S.A.), "Fan Engineering," by Willis H. Carrier (Buffalo Forge Co., U.S.A.), "Power Heating and Ventilating," by C. L. Hubbard, B.S., M.E. (McGraw and Hill Book Co., U.S.A.), "Heating and Ventilation" (Sturtevant Engineering Series), "Air Currents

and the Laws of Ventilation," by W. N. Shaw, D.Sc., F.R.S. (University Press), and "Heating Systems," by F. W. Raynes (Longmans & Co.), to each of which, to The American Blower Co., to The National Radiator Co., Ltd., to Messrs. J. F. Philips & Co. (owners of the copyright of Mr. Barker's book), and to many other authorities he has to express his indebtedness for much information and for many illustrations, tables, &c., reprinted herein.

While every endeavour has been made to avoid omissions and errors, the writer, who returned again to active service before the work was completed, will welcome any corrections or suggestions for future editions.

In conclusion he has to thank sincerely his old friend Mr. A. F. Browne, A.M.I.Mech.E., for the valuable assistance he has rendered by reading of proofs and checking of figured examples.

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TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF TABLES

LIST OF EXAMPLES

SOME MODERN METHODS OF VENTILATION

WITH SPECIAL REFERENCE TO PUBLIC BUILDINGS

CHAPTER I

THE GENERAL PRINCIPLES OF VENTILATION

1. Definition of Ventilation.

Any room or building used for the habitation of human beings should be provided with a plentiful supply of pure air.

Ventilation is defined as the art of passing a predetermined volume of *conditioned* air into the space to be ventilated, and out of it again, and the proper distribution of the air during its passage through the ventilated space.

2. Requirements of Ventilation.

Ventilating systems are installed to :—

(*a*) Maintain certain predetermined standards of purity, temperature, and humidity within the apartment to be ventilated ;

(*b*) To remove from and prevent the formation therein of objectionable or insanitary odours.

Although the reason for the necessity of ventilation is not very clear, it is an accepted fact that it is more or less injurious to any animal to breathe air a considerable part of which has already been expired, and there can be no doubt

that loss of health and increased liability to certain disease is associated with living in such air.

Questions of personal comfort as regards temperature, odour, and perspiration also form indisputable arguments in favour of efficient ventilation, apart altogether from the hygienic aspect of the matter.

3. Standards of Ventilation.

Many of the existing standards of ventilation have been founded on the belief that carbon dioxide was the dangerous element in expired air.

The requirements of ventilation as to air purity are more or less arbitrary, and no rational standard has yet been fixed; but it has been clearly and definitely proved by the recent magnificent research work of Dr. Leonard Hill, of the University of London, and of others, that *the presence of an increased quantity of carbon dioxide* (CO_2) *in respired air, even up to* 300 *parts as against the previously accepted limits of* 6 *to* 8 *parts in* 10,000 *parts of air, has no deleterious effect,* and does not affect to the slightest degree the concentration of carbon dioxide in the blood. Therefore it would appear that carbon dioxide is interesting only as indicating how much respiration the air has undergone, and in this way it serves as an index of the contamination of the air with organic impurities from the lungs and bodies of the occupants.

Even more disconcerting to the old school of hygienists must be the growing practice of American engineers to arrange for recirculation of the air many times in succession, the washing plant, together with the addition of a very small percentage of fresh air, being relied upon to maintain the atmosphere of the ventilated apartment in a pure condition. Needless to say, this system effects considerable economy in the fuel bill in cold weather.

In a paper before the American School Hygiene Association, an architect said, referring to the system of recirculated air:—

"Such a system has been installed in the new Wetherfield School, Hartford, capable of *operating with only* 10 *per cent. of outdoor air.*

If this method is a success, mechanical ventilation will indeed have reached its logical, if somewhat startling, conclusion." (*Heating and Ventilating Magazine*, September, 1915.)

Dr. Leonard Hill, at a meeting of the Institution of Heating and Ventilating Engineers, October, 1914 stated :—

"Now I want to maintain with the utmost of my power that the whole question of ventilation which matters is this: the effect of heating and ventilating not upon the chemical purity of the atmosphere but upon the skin. *The physical effect upon the skin is the supremely important thing for us, not its chemical effect.*"

Essential features of the atmospheric condition of the ventilated apartment are :—

(*a*) Equable temperature, 60° to 65° Fahr.

(*b*) Moderate relative humidity or moisture contents, 45 to 60 per cent.

(*c*) Air movement which, though not sufficient to be felt as objectionable draught, yet continually removes and replaces the vitiated air, surrounding the body with fresh air, thus assisting the natural action of the pores of the skin and stimulating the nerve tissues (velocity of 1 to 3 feet per second).[1]

(*d*) Clean air, *i.e.*, free from the suspended impurities prevalent in large cities and near busy roads—dust, insects, oily vapours from gasoline engines, soot, etc.

(*e*) Freedom from insanitary or objectionable odour.

In rooms where the glass and wall exposure is considerable, *ventilation for the removal of bodily heat* need not be considered except when the apartment is to be artificially cooled. *In crowded audience halls, and even in schoolrooms, it is the determining factor.*

The minimum quantity of air to be introduced into any apartment for the purposes of efficient ventilation is a matter calling for considerable experience and careful consideration

[1] An interesting development of the ventilating theories propounded by Dr. Hill is Marlowe's "Normair" process (Pulsometer Engineering Co.), used to a considerable extent in drying installations in which, by the use of a patented alternator, the fan duty is varied from zero to maximum output two or three times per minute, thereby delivering the air in gusts, thus avoiding monotony and more nearly imitating Nature's bracing breezes.

of the local conditions. For instance, a cathedral having about 7 feet super floor area per seat, a period of continuous occupation of about one and a quarter hours, and no smoke difficulties to contend with, must be considered from an entirely different standpoint to, say, a cinematograph theatre having a floor area of possibly 4 feet super per seat, continually occupied for ten to twelve hours per day, smoking being allowed.

TABLE I.—**Quantity of Air Required for Adequate Ventilation.**

(Based on Temperature Rise due to Bodily Heat.)

Cub. ft.-hr.-person.	Type of Building.	Temp. Increase in deg. Fahr. due to Bodily Heat (Excluding Transmission and other Losses).	Total Parts of CO_2 in 10,000 Parts of Air (External Air, 4 Parts).
6,000	Hospitals	2·78	5·0
5,000		3·30	5·2
4,000	—	4·17	5·5
3,000	Music-halls, and cinematograph theatres (smoking).	5·55	6·0
2,500		6·68	6·4
2,000	Schools, theatres, and audience halls (no smoking allowed).	8·35	7·0
1,800		9·25	7·3
1,500	—	10·1	8·0
1,200	Churches	13·9	9·0
1,000	—	16·7	10·0
750	—	22·2	12·0

Another, although less scientific, method of ascertaining the quantity of air required in audience halls, etc., is based on a definite number of air changes per hour, the cubic contents of the apartment being known.

TABLE II.—**Quantity of Air Required for Adequate Ventilation.**

(Based on Changes per Hour: Thompson.)

Public waiting-rooms	4 changes.	Textile mills	4 changes
Public toilet-rooms	10 ,,	Foundries	3 ,,
Locker-rooms	6 ,,	Hotel lobbies	4 ,,
Small meeting-halls	4 ,,	Hotel kitchens	15 ,,
Public offices	5 ,,	Engine-rooms (exclusive of engine heat losses)	6 ,,
Private offices	4 ,,	Laundries	20 ,,
Ball-rooms	4 ,,	Public libraries	3 ,,
Public dining-halls	4 ,,	Private libraries	4 ,,
Banquet-halls	5 ,,		
Railroad round-houses	12 ,,		

Natural air changes without mechanical aid, due to doors, windows, fire places, etc., average one to two per hour for well-built apartments, and two to three for medium quality do.

Generally, it is found undesirable to blow air into small kitchens, lavatories, etc., but to rely solely on mechanical extraction, in order to avoid distributing objectional or insanitary odours. For the same reason, in large apartments of the type mentioned the extraction system should be designed to remove 20 per cent. more air than the plenum [1] system supplies, in order to maintain a slight vacuum, which will overcome any natural aspirating tendencies.

4. Sources of Vitiation.

Sources of vitiation which may have to be considered, either in conjunction with or separately from the foregoing factors, before a final determination of the total quantity of air required for ventilation purposes can be made are :—

(a) *Human.*

An adult man gives off 1,000 grains of water vapour and 0·6 cubic feet of carbon dioxide (CO_2) per hour,

[1] "Plenum" system is the name given to the fresh air supply when forced into the apartment under slight pressure by means of a centrifugal fan.

which is equivalent to an increase of from 4 to 5 parts of CO_2 per 10,000 parts of air.

Also an adult in repose under average conditions emits 350 British thermal units[1] (B.Th.U) of bodily heat per hour, but if at heavy manual labour this will be increased to 550 B.Th.U. per hour.

(b) *Machinery.*

Each horse-power utilised in overcoming bearing, belt, etc., friction produces 2,530 B.Th.U. per hour, and each kelvin or Board of Trade " unit " of electricity 3,400 B.Th.U.

Example 1.—An electric motor or dynamo of 500 kilowatt capacity, and, for round figures, say 80 per cent. efficiency, is placed in a closed brick chamber which has to be ventilated to

Table III.—**Efficiency of some Gas and Electric Consuming Apparatus.**

	Per cent.
Electric transformers (static)	85 to 98
,, converters (alternating to direct)	80 to 92
,, motors	65 to 85
,, cookers	60 to 85
,, arc lamps (Molesworth)	19·0
,, metal filament lamp (Molesworth)	15·0
,, carbon filament (Molesworth)	5·0
Gas cookers	34·0
,, engines	25 to 32
,, incandescent mantle (Molesworth)	1·88
,, ordinary flame (Molesworth)	0·4

maintain the temperature constant. Ascertain heat units produced by the machine at full load.

Obviously out of each 100 kilowatts only 80 are usefully converted, the remaining 20 per cent., or $5 \times 20 = 100$ k.w., are transformed into heat, and produce $3{,}400 \times 100 = 340{,}000$ B.Th.U. per hour.

Example 2.—An electric resistance connected in an arc lamp circuit absorbs 170 volts when passing 10 amps. *i.e.*, 1·70 kilo-

[1] A British thermal unit is defined as that amount of heat which will warm 1 lb. of water through 1° Fahr. Strictly speaking from 32° to 33° Fahr.

watts. Therefore the heat generated per hour equals 1·7 × 3,400 = 5,780 B.Th.U. per hour.

EXAMPLE 3.—An incandescent gas-burner consumes 3·5 cubic feet of 550 B.Th.U. gas (the usual heat value of town gas). The light efficiency of such a burner is approximately 1 per cent., which is practically negligible as regards our calculation. Therefore the heat generated will be

$$550 \times 3{\cdot}5 = 1{,}925 \text{ B.Th.U. per hour.}$$

Some common efficiencies of electric and gas consuming appliances are given in Table III.

(c) *Sun Effect.*

Unclouded sun shining on 13-inch brick wall transmits 6·0 B.Th.U. per sq. feet per hour, and on a glass surface 150 B.Th.U. per hour.

EXAMPLE 4.—Hence a photographic studio or similar building having a roof of glass area 100 feet by 200 feet or 20,000 feet super., will require 20,000 × 150, or 3,000,000 B.Th.U. removed per hour, in order to avoid a rapid rise of temperature.

5. American Ventilating Legislation ("Heating and Ventilating Magazine").

Many American states have realised the vital importance of ventilation as affecting public health and have enacted laws regarding the heating and ventilation of schoolhouses, audience halls, etc., further specifying tests to be made to prove that the provisions of the enactments have been duly carried out.

Indiana (Chap. 144, Acts 1909, Sect. 6).

"Heating and ventilating systems of all kinds shall take fresh air from outside the school buildings, evenly diffuse the same throughout each schoolroom during school session and withdraw foul air from said schoolroom at a minimum rate of 1,800 cubic ft.-hr. for each 225 cubic feet of said schoolroom space regardless of outside weather conditions."

Ohio (Sect. 30).

"All other assembly halls and theatre auditoriums shall be heated and ventilated by a system which will supply to each auditor not less than 1,200 cubic feet of air per hour."

"No floor register for heating or ventilating shall be placed in any aisle or passageway."

Cleveland (Ordinance 29,798, October, 1913).

"Every hall, auditorium, or room of every building hereafter erected or converted to use as a schoolhouse, theatre or place of public assembly or entertainment, shall have in continuous operation, while occupied, a system of ventilation so designed and installed as to provide not less than 25 cubic feet of outer air per minute — 1,500 feet per hour—for each person for whom seating accommodation is provided."

Similar laws are either being enacted or are in force in Massachusetts, Minnesota, Tennessee, Florida, Louisiana, Vermont, etc.

6. Law of the Air Circuit.

For ventilation purposes in the British Isles the minimum outside air temperature is assumed at 30° Fahr. and the usual indoor temperature at 60° Fahr.; hence 65° Fahr. is usually the maximum temperature required in air conditioning (unless some form of hot blast heating, which is outside the scope of the present work, is adopted), thus allowing for a 5° Fahr. drop in the ducts and other air passages. If greater drop in ducts is anticipated, then due allowance must be made.

Further, the average temperature in this country during the winter months is 41° Fahr., so that the temperature increase required except during the prevalence of what may be described as severe weather is only 24°, corresponding to an increase of volume of 5 per cent. (approximately), so that in order to simplify figuring we shall regard the volume of air as constant, and make no provision for a volumetric variation due to temperature in subsequent calculations.

The dominant physical law of the ventilated space is that of convection, and, while it is not convenient to express it in any mathematical form, yet it is at once the condition of success and the cause of many failures. A person warmer than the air in the immediate neighbourhood causes an upward air current; every surface colder than the air in contact with it causes a downward current. Convection currents are sometimes termed aspiration currents, and in ventilating high buildings special steps must be taken to

avoid this fault by means of doors, etc., or staircases, enclosed lift shafts, etc.

7. Composition of Air.

Air is a mechanical mixture of nitrogen 21 parts and oxygen 79 parts by volume, containing 0 to 4 per cent. of moisture and 3 to 30—usually 4—parts of carbon dioxide per 10,000 parts of air.

At 30° Fahr. dry air weighs 0·081 lb. per cubic foot and 1 B.Th.U. raises 51·1 cubic feet by 1° Fahr., but at 65° Fahr. the weight is reduced to 0·75 lb. per cubic foot and the quantity increased to 54·6 cubic feet.

8. Tempering of Air.

As stated in par. 6, we are only concerned in the present work with the heating of air from a minimum temperature of 30° Fahr. to a final temperature of 65° Fahr., and (approximately) 1 B.Th.U. will raise 53 cubic feet of air at the average temperature of 47·5° Fahr. through 1° Fahr.

Example 5.—To raise 1,000,000 cubic feet of air through 35° Fahr. (neglecting humidification) the heat required is

$$\frac{35 \times 1{,}000{,}000}{53} = 660{,}000 \text{ B.Th.U.}$$

Example 6.—Referring to Example 1, par. 4, we obtained a value of 340,000 B.Th.U. per hour as the heat to be removed if the temperature is to be maintained constant. To reduce the quantity we will allow a rise of 10° Fahr., which will not cause discomfort, and we then get

$$\frac{53 \times 340{,}000}{10} = 1\ 800{,}000,$$

approximately 1,800,000 cubic feet of air per hour, or 30,000 cubic feet of air per minute (neglecting heat transmission through walls, floors, ceiling, etc.).

Conversely, 1 B.Th.U. is liberated by each 53 cubic feet of air for each degree Fahrenheit through which it is allowed to cool.

Example 7.—The heat losses of an apartment are 1,000,000 B.Th.U. per hour, and the temperature required is 60° Fahr. Ascertain quantity of air to be introduced per minute at

a temperature of 120° Fahr. (neglecting ventilation and humidity) :—

Temperature drop available = 120° to 60° = 60° Fahr.

$$\text{Cubic feet of air per minute} = \frac{1{,}000{,}000 \times 53}{60 \times 60} = 14{,}752\text{—say}$$

15,000 cubic feet per minute.

9. Humidity of Air.

By the term " humidity " is meant the quantity of water vapour or moisture mixed with a given volume of air.

" Relative humidity " is the term given to the ratio of the weight of water vapour contained in a given volume of air to the weight which this same volume would contain when fully saturated at the same temperature ; it is expressed as a percentage.

The difficulty, if any, experienced in grasping the details of the phenomenon connected with relative humidity is due to the fact that *the capacity of air to absorb moisture increases with rise of temperature*, thus :—

Table IV.—**Increased Capacity of Air for Water Vapour with Increase of Temperature.**

Temp. of Air (deg. Fahr.) .	30	40	50	60	70	80	90	100
Maximum weight of water absorbed in grains per cubic foot . . .	1·94	2·85	4·08	5·75	7·98	10·9	14·7	19·7

Hence 1 cubic foot of air containing 5·75 grains is saturated, *i.e.*, " full up," or can carry no more water vapour at an air temperature of 60° Fahr. ; but if the temperature of the air be raised to 90° Fahr. the relative humidity is automatically reduced to 40 per cent., the *actual weight of water vapour present remaining constant.* It should be noted that the fall in the relative humidity is due entirely to the *increased capacity* of the air to hold a greater quantity of water vapour (*i.e.*, 19·7 grains for saturation, or 100 per cent. relative humidity) at the higher temperature.

10. Cooling by Evaporation of Water.

It is a well-known fact that when water evaporates, or is converted into the gaseous state, heat is absorbed or rendered latent. Common applications of this phenomenon are seen in the soldier's felt-covered water bottle carried on service, the unglazed earthenware water bottle commonly used in the East, and the camel bags made of canvas, which also carry water. In the former the felt is soaked in water, and in the latter the contained liquid slowly percolates through the material of which the vessel is made, the result in each case being that the liquid on the outside of the container slowly evaporates by absorbing the heat necessary for the process, chiefly by conduction, from the contents, resulting in a cool drink. The refrigerating or cooling effect as applied to ventilation is $8\frac{1}{2}°$ Fahr. for each grain of moisture evaporated per cubic foot of air, at ordinary temperatures.

11. Dew Point.

Since the capacity of air to absorb moisture increases with rise of temperature, the converse must obviously be true—viz., as the temperature of saturated air is lowered, the excess moisture must be deposited when the dew point or saturation temperature is reached. Consider a cubic foot of air containing 2·85 grains of water vapour :—

TABLE V.—**Increase in Relative Humidity due to Reduction of Temperature.**

(Actual Weight of Water present being Constant.)

Temp. (degs. Fahr.).	90	80	70	60	50	40
Relative humidity .	19	26	36	48	67	100

It is clear, therefore, that for a moisture contents of 2·85 grains per cubic foot, 40° Fahr. is the dew point or temperature of saturation, and if cooling is continued the excess

moisture will condense out in the form of water drops or "rain." This effect is clearly seen on glass windows in cold weather, in which the air of the room is cooled by contact with the glass to a temperature below the dew point, when the excess moisture condenses out—*i.e.*, the glass "steams."

Again, a tumbler containing an iced drink is frequently covered on the outside with water drops, which have condensed out of the air in contact with the glass, which is therefore at a temperature below the dew point of the air contained in the room.

12. Effect of Humidity on Ventilation.

Under normal conditions the air has a relative humidity varying from 35 per cent. on dry days to 98 per cent. (*i.e.*, within 2 per cent. of the dew or saturation point) on wet, misty days ; but for health and comfort a room temperature of 60° Fahr. and a relative humidity maintained between the limits of 45 per cent. to 60 per cent. of saturation, corresponding to an average moisture contents of 2·9 grains per cubic foot of air at that temperature, is found to give the best results.

The humidity of the atmosphere exercises a strong influence over bodily sensations of heat and cold, and is of great importance in breathing, as the quantity of water vapour exhaled with each respiration is much greater when the air is dry than when it is damp. The secretion from the skin is increased when the air is dry, but then there is very little perspiration, because it is at once converted into vapour by the dry air instead of condensing in drops of sweat upon the surface of the body.

The (latent) heat required for the conversion of the perspiration to the gaseous form is abstracted principally from the body, hence the feeling of *coolness* produced by standing in a current of *warm*, dry air. In such an atmosphere much of the perspiration is unnoticed, but in a hot, humid (*i.e.*, moist) atmosphere it is readily perceived.

Cold as well as heat is much more readily borne by the human subject when the air is dry.

To use the chart:—

The ordinate of the curve shows the amount of water, in grains of moisture, contained in one cubic foot of air when the temperature of the dry bulb is represented by the abscissæ, for any degree of saturation. It also shows what the wet bulb will be for any given degree of saturation.

(a) Given a dry bulb temperature of 64° Fahr., and a relative humidity of 50 per cent.,

Required to find the corresponding wet bulb temperature:—

Follow the vertical line representing 64° Fahr. upwards until it strikes the curved line representing 50 per cent. saturation, and note the position of this point on the line sloping downwards from left to right, which is approximately 53·5° Fahr.

(b) Given the dry bulb reading of 60° Fahr., and the wet bulb 56° Fahr.,

Required to find the relative humidity:—

Follow the vertical line representing 60° Fahr. until it strikes the sloping line representing 56° Fahr., and note the position of this point of the curved line, which is 80 per cent.

(c) Given the dry bulb reading of 32° Fahr. and a relative humidity of 100 per cent. (*i.e.* saturation),

Required to find the relative humidity of the air when its temperature is raised to 60° Fahr. without the addition of moisture:—

Follow the vertical line representing 32° Fahr. until it strikes the curve representing 100 per cent. relative humidity (saturation). Follow the horizontal line running from left to right through this point until it strikes the vertical line representing 60° Fahr. and note the position of this point on the curved line which gives 37 per cent. relative humidity.

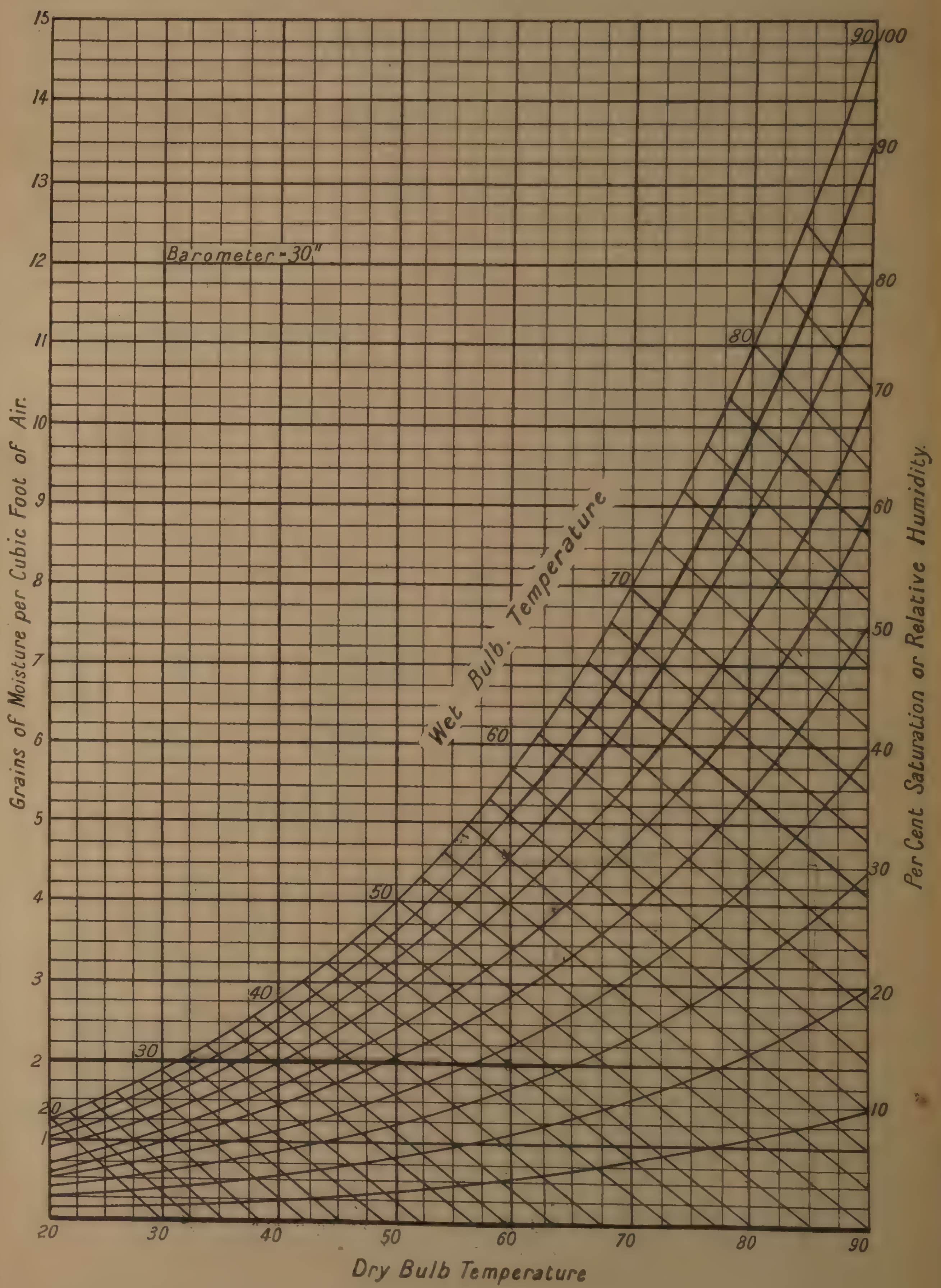
(d) Given the dry bulb reading of 85° Fahr. and a relative humidity of 33 per cent.,

Required the wet bulb temperature, i.e., the maximum possible cooling effect (saturation):—

Follow the vertical line representing 85° Fahr. until it strikes the curved line representing 33 per cent. relative humidity, and note the position of this point on the sloping line which gives the wet bulb temperature of 65°.

Therefore 85° — 65° = 20 degrees reduction, the air being saturated, *i.e.*, 100 per cent. relative humidity.

TABLE VI.—**Psychometric Chart.** (By the courtesy of The Buffalo Forge Co., U.S.A.)



13. Cooling by Mechanical Refrigeration.

Cooling large volumes of air by means of refrigerating plant or ice is commercially out of the question.

EXAMPLE 8.—Consider an installation requiring only 20,000 cubic feet of air per minute to be reduced by 15° Fahr. (neglecting the heat to be dealt with if the air be reduced to a temperature below the dew point).

$$\frac{20\,000 \times 60 \times 15}{53} = 340{,}000 \text{ B.Th.U. of refrigeration required per hour.}$$

The latent heat of ice (*i.e.*, the heat absorbed when one pound of ice at 32° Fahr. is completely transformed to the liquid state at the same temperature) is 142 B.Th.U., or per ton of refrigeration $142 \times 2{,}240 = 318{,}080$ B.Th.U. against 29,120,000 B.Th.U. contained in one ton of fuel.

H. R. Kempe, in "The Engineer's Year Book, 1914," puts the power consumption at from 50 to 60 kilowatt hours per ton of refrigeration produced by a motor-driven ammonia compression plant.

If refrigeration is obtained in the form of ice it is rather worse, as the handling and transport alone of $1\frac{1}{4}$ tons of ice per hour required for a 20,000 cubic feet per minute plant is a financial consideration.

If the air is cooled below the dew point there would be the additional heat units, corresponding to $8\frac{1}{2}$° Fahr. per grain of moisture deposited per cubic foot of air cooled, to be removed. Also a further increase of temperature, with a corresponding expenditure of fuel, would be required to reduce the relative humidity from 100 per cent. (saturation) to something of the order of 50 to 65 per cent.

CHAPTER II

THE APPARATUS EMPLOYED

1. System to be Adopted.

It is obvious from a consideration of the matter contained in the previous chapter that the air supplied for any first-class ventilating system must be efficiently conditioned, *i.e.*, warmed or cooled and humidified as required, before being passed into the apartment to be ventilated. This fundamental condition leaves us, therefore, with practically the choice of two systems only :—

(*a*) Mechanical Plenum and Natural Exhaust.

(*b*) Mechanical Plenum and Mechanical Exhaust.

Although there are certain cases where the natural aspirating tendencies, due to position of site, height of building, etc., may cause a Mechanical Plenum and Natural Exhaust System to work satisfactorily, yet there is always grave risk, in dealing with the large volumes of air required, of objectionable outdoor draughts through doors, passages, etc., connected with the ventilated apartment, so that in first-class work the second system (*i.e.*, Mechanical Plenum and Mechanical Exhaust) is almost invariably adopted. It also has the additional advantage, which will be appreciated by architects and others interested in the decorative scheme, that when this system is adopted the extraction registers are materially reduced in size.

This system is often termed the Balanced System,[1] one fan—the plenum—being used to " pump " the air in under a slight pressure, sufficient to overcome the friction of ducts,

[1] The writer has been able to balance so perfectly with this system, that a variation of one point on the extraction fan speed regulator, with the plenum fan running at rated output of 1,200,000 cubic feet of air per hour, will convert an " inwards " into an " outwards " draught, as shown by the revolutions of an anemometer held in the main doorway.

heater, washer, register grids, etc., etc., and a second fan—termed the extractor or exhauster—being used for the removal of the vitiated air. This latter fan may or may not be called upon to set up a pressure, depending as to whether much friction is to be expected from the ducts used to connect the extraction registers to the fan and finally to the external air.

In this system it is usual to design the plenum fan to introduce a slightly larger quantity of air (15 to 25 per cent.) than is being removed by the extraction fan, as this keeps the building under a slight pressure, thereby preventing the ingress of unconditioned air, *i.e.*, foggy air and cold draughts.

For audience halls and apartments of similar dimensions the problems of efficient air distribution and the avoidance of draughts are greatly increased owing to the large dimensions of such buildings, and these are rendered still more difficult if floor gratings are impracticable, which will be the case if seats are movable. In any case, floor gratings are usually undesirable from a sanitary point of view unless arranged in step risers or chair frames.

The Balanced System may be still further sub-divided as follows :—

(*a*) Upward ventilation.

(*b*) Downward ventilation.

In the former, which is the scheme it is proposed to discuss in detail, the plenum air is introduced through wall registers at a height not exceeding 8 feet above the floor level and extracted through registers fixed in the ceiling. Thus the air flows upward in accordance with natural currents induced by the heat of the body and the breath, the vitiated air being carried away and the incoming air uncontaminated. It facilitates distribution if the air is introduced at the same or a slightly lower temperature (1° to 2° Fahr.) than that maintained in the apartment, so that the air falls, rather than rises, on entry.

The theory that CO_2 is heavier than air and therefore falls, remaining in a layer at the floor level, is not borne out in

practice, which is probably due to the upward convection currents, caused by bodily heat.

In the downward method the air is admitted through registers in the ceiling, or several feet above the floor level, and removed through vent registers in the walls at the floor line. Perfectly designed by a ventilating engineer of wide experience, provided with perfectly figured and constructed ducts which are necessarily of an even more elaborate nature than used in the upward system, and operated by a thoroughly competent and intelligent engineer, equally good results are obtained with this system.

2. Position of Plenum Intake.

For the intake of the fresh air, a position as free from contamination, and facing north if possible, should be selected, smoky chimneys, sanitary vent pipes, and the discharge from the mechanical extraction system being avoided, unless it is decided to experiment with re-circulated air (as mentioned in Chapter I. par. 3), in which case provision must be made for connecting to the mechanical extraction system, at a convenient point, by means of a by-pass duct.

The opening for the intake should either be fitted with louvred boards or iron bars, to prevent unauthorised entry to the building, and in either case expanded metal or wire netting is added to keep out birds, straw, rag, and other débris (*i.e.*, coarse filtering).

Incidentally this duct, if fitted with a suitable ceiling bolt and shackle, may in certain instances prove a convenient route for getting the fan, boiler, motor, washer, and other bulky gear into position.

3. Position of Plenum Apparatus.

The plenum apparatus, which has considerable weight, should, wherever possible, be placed in the basement and connected to the main intake opening by means of a duct built in white glazed brick, drains fitted with adequate traps being provided as required to facilitate cleaning

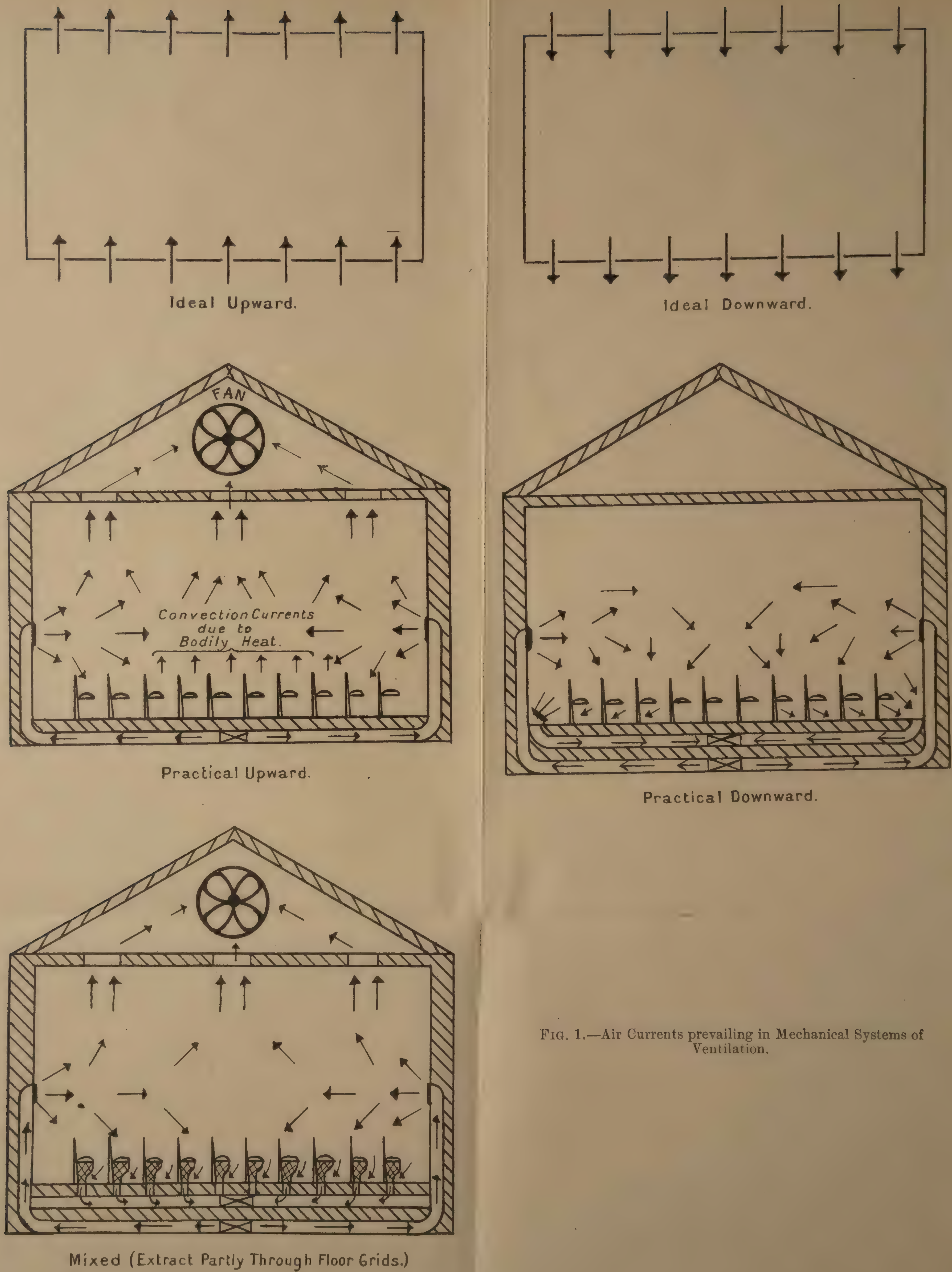


FIG. 1.—Air Currents prevailing in Mechanical Systems of Ventilation.

and to deal with rain or melting snow, which may be drawn in during the prevalence of high winds and under the

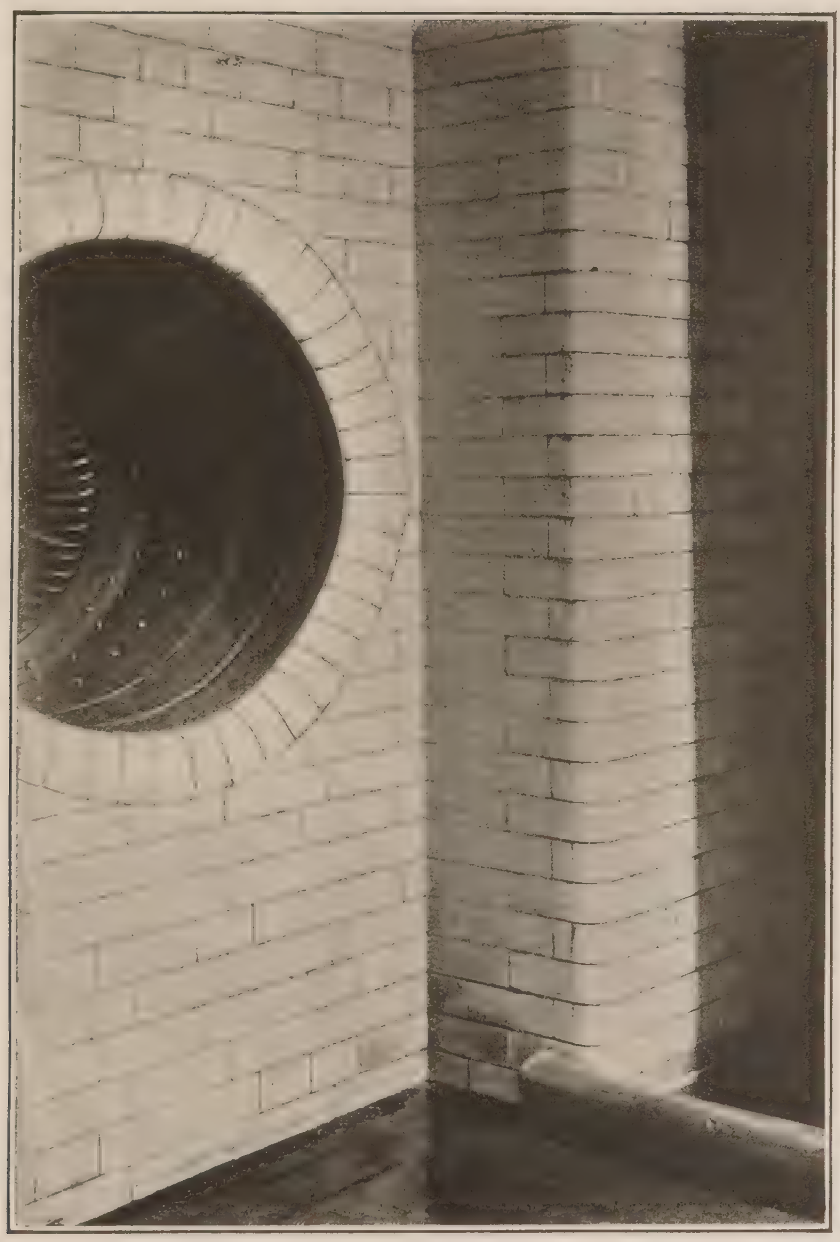


FIG. 2.—Glazed Brickwork to Inlet of Centrifugal Fan.
(By the courtesy of Messrs. Davidson & Co., Ltd.)

influence of the appreciable suction which will exist in the vicinity of the intake if the dimensions are at all restricted.

By adopting this scheme the prospects of noise troubles are considerably diminished. Also the air traversing masonry or concrete ducts will be delivered to the ventilated

apartment at a more equable temperature in summer and in winter than if designed to traverse long uninsulated ducts fixed under the slates forming the roof of the building.

4. Arrangement of Plenum Apparatus.

Any complete system must include an air-washing and humidifying plant to remove dust and dirt from the air; and in order to regulate the degree of humidification it is essential to provide heating surface at the inlet to the washer. There is a drop in the temperature of the air during its passage through the spray chamber, so that a second heater is required at the discharge of the washer.

Obviously it is impossible to rely on natural means for circulating the large volumes of air necessary for the efficient ventilation of public buildings, audience halls, etc., and the most convenient air mover is the electrically driven multi-blade centrifugal cased type fan, discussed in Chapter V.

The position of the fan in relation to the washer, and the best relative arrangement of the gear, is a matter for special consideration in each individual case, but it should be borne in mind that the "draw-through" (washer and heater on suction side) is slightly more satisfactory than the "blow-through" design (washer and heater on discharge side), as the velocity of the air can be gradually increased up to the fan and the excessive portion of the velocity head of the fan converted to pressure head.[1]

5. Distribution Systems.

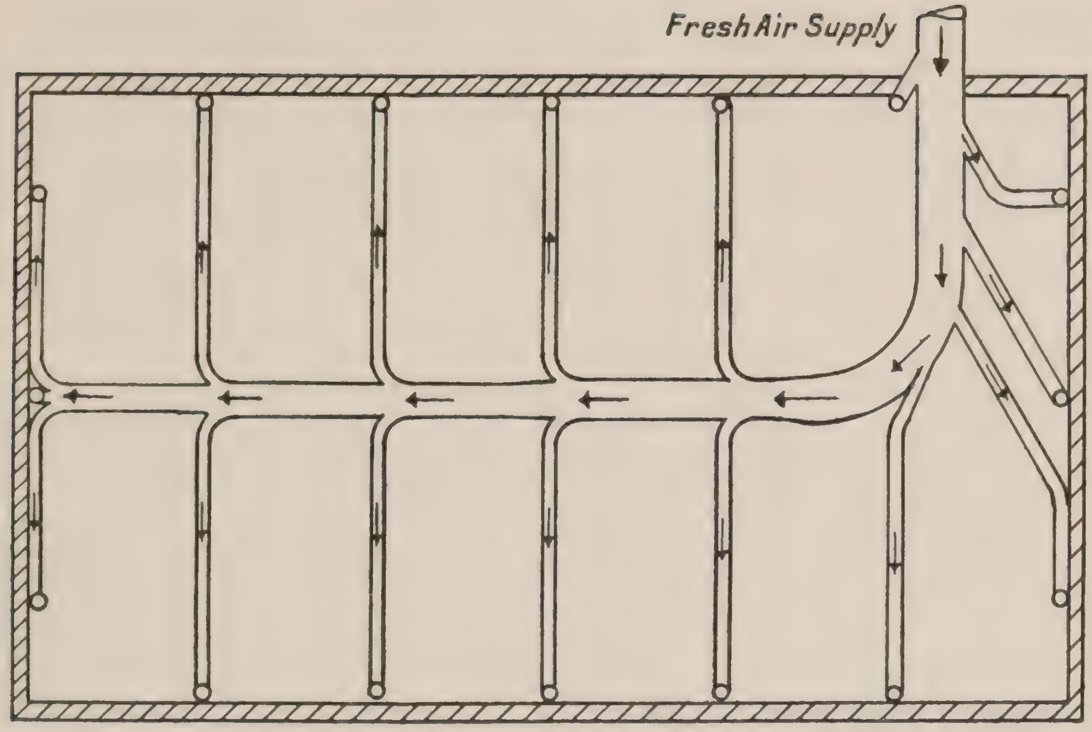
The method of obtaining a supply of conditioned air having now been explained, there remains the distribution system to consider. This can be divided into two types, *i.e.* :—

(*a*) Main Trunk System.

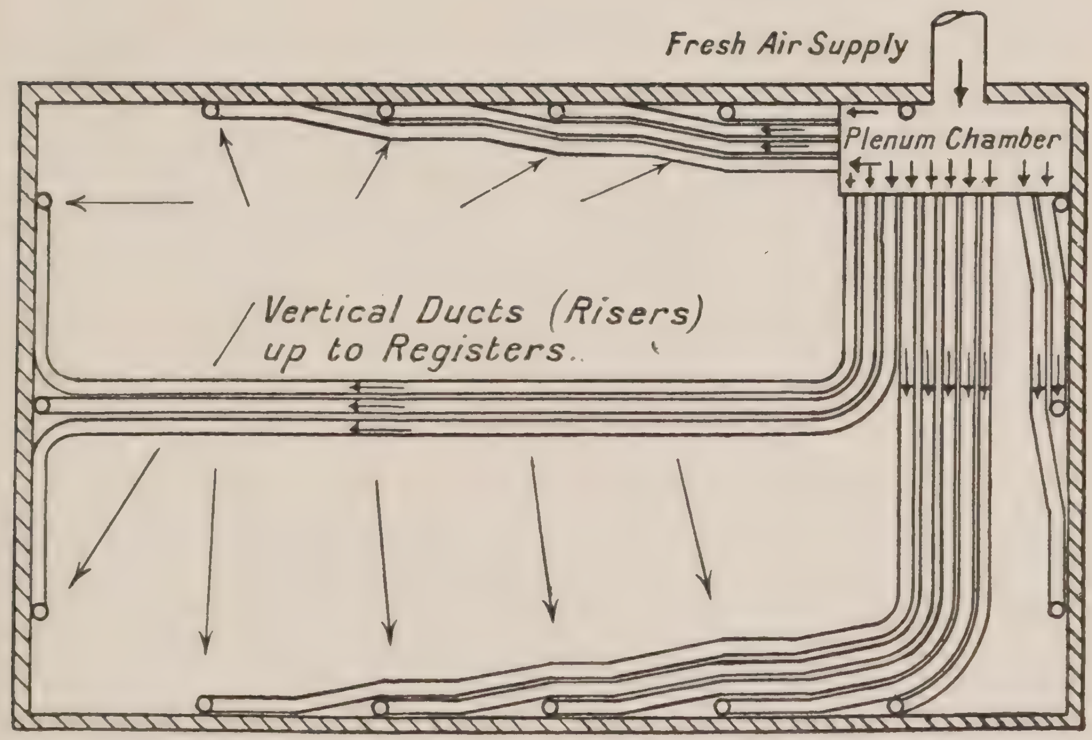
(*b*) The Plenum Chamber System.

In the Main Trunk System, which can be roughly compared with the tree system of electrical wiring, a main supply trunk is run through the centre of the area to be ventilated

[1] These details are more fully discussed in Chapter III.



Main Trunk System of Ducts.



Plenum Chamber System of Ducts.

FIG. 3.—Main Trunk and Plenum Chamber Systems of Air Distribution.

and branches are taken off, being further sub-divided as required. This is the arrangement commonly used when the fan is to supply air for ventilation purposes only, and the one herein recommended.

When the hot-blast system is adopted, and the fan system is used for heating as well as ventilation purposes, the Plenum Chamber System, which is analogous to the distribution board system of electric wiring, is adopted. In this design the system is sub-divided into a large number of smaller systems, the feeder trunks from which are all collected in a main plenum chamber, which is so arranged, by means of by-pass dampers, that all cold, all hot, or air at any intermediate temperature, can be delivered to any duct, the said dampers frequently being automatically controlled by thermostats placed in the apartment served by the duct. It will be observed that this system has special advantages when temperature control of the conditioned air over considerable range is desired in a number of apartments.

Summarising the details of the system advocated :—

Cold air is drawn in, preferably from the top of the building or from some area remote from sources of contamination, through a protected opening in the wall, and down a shaft built for that purpose in glazed brick, to the basement. There it is drawn through a tempering coil, which warms it to the required degree for humidification, passes through the washer, where it is scrubbed and humidified to the required degree, and so on to the re-heating coils for final heating. The air is then drawn into the fan, from which it is discharged at the calculated velocity and at a pressure sufficient to overcome the friction loss of the complete duct system, which distributes it as required to the various parts of the building, through the registers, and into the apartments to be ventilated.

6. Position of Plenum Registers.

Every ventilating engineer holds more or less decided views on this point, but, for general work, positions in the walls at a height of 6 ft. 6 in. from the floor line to the

lower edge of the register are an arrangement which gives very satisfactory results if the air is delivered in a horizontal direction, and at a maximum velocity, through the *free area* of the grating at 200 to 300 feet per minute for the smaller rooms, and at 400 to 500 feet per minute through

FIG. 4.—Plenum and Extraction Registers.

the free area [1] for large halls, in order to attain a greater distance.

7. Position of Extraction Registers and Fan.

As previously indicated, the present writer prefers to place the extraction registers in the ceiling.

If a false roof or ceiling space can be provided immediately above the ventilated apartment, this simplifies the design enormously, as ducts then become unnecessary, the whole

[1] See Chapter II., par. 9.

of the space being made more or less air-tight and the vitiated air extracted from the apartment *viâ* the ceiling registers by a simple and cheap ring type of fan, which can be bolted direct to the wall.

When it is necessary to draw vitiated air from a large number of smaller apartments, the various registers are connected to convenient extraction ducts, rising up through the different floors, being in turn collected in the attic or other place forming the exhaust chamber and connected to the extraction fan, which, in an instance of this kind, would probably be of the double inlet multiblade cased centrifugal type, in order to deal with the resistance set up to the air flow.

Systems containing ducts having high and low resistance values cannot be connected to the same exhaust chamber, but to separate fans of suitable design, all the ducts of one system being figured for equal resistance values when dealing with the required volume of air, as discussed in Chapter IV., par. 5, in order to obviate subsequent regulation by the fixing of permanent obstructions, which are of course highly inefficient.

TABLE VII.—**Wind Velocities and Equivalent Velocity Pressures.**

Velocity in feet per minute .	4,850	4,400	(aver.) 3,740	(aver.) 2,860	1,760	1,320	880	264
Do., miles per hour . .	55	50	45—50	35—30	20	15	10	3
Description .	great storm	storm	very high wind	high wind	brisk gale	pleasant breeze	—	just perceptible
Equivalent velocity in inches of water column .	1·5	1·2	0·9	0·5	0·2	0·1	0·05	—

8. Extraction Fan Exhaust.

If possible a sheltered spot, where high winds will have little or no effect in causing back pressure on the fan, should be selected for the extraction fan exhaust, as, otherwise,

FIG. 5.—Extraction Fan Exhaust, for a duty of 2½ million cubic feet per hour.

excessive variations in the performance of the fan will be observed.

If no convenient and sheltered area is available, then a tower form of exhaust having louvred sides may be built on the roof so that one point of the compass is always favourable to the discharge.

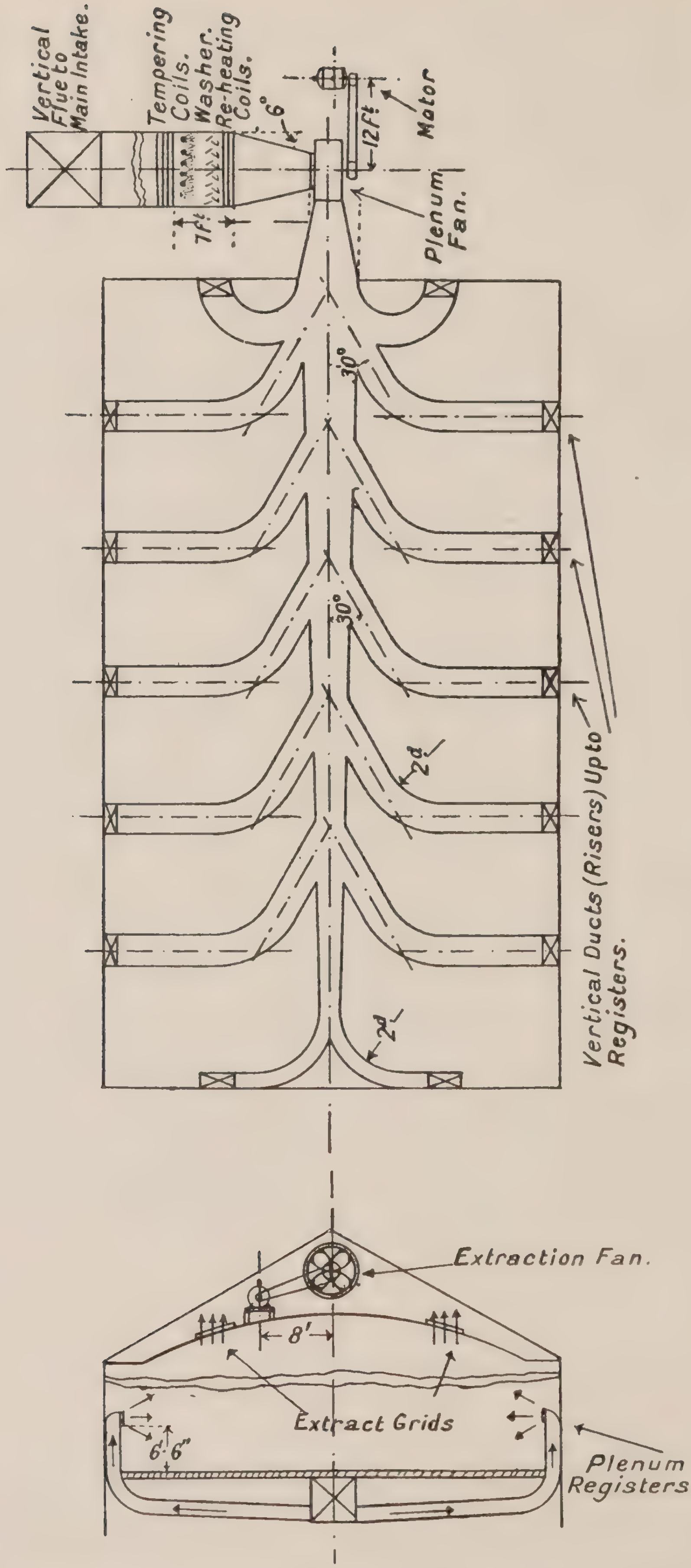


FIG. 6.—Model Design of Ventilating Installation on the Balanced System.

For smaller installations revolving cowls prove very useful. As in the case of plenum intake ducts, provision should be made for draining rain and melting snow away, which will drift in when the fan is not moving unless automatic shutters are fitted.

The question of back draught becomes of special importance in dealing with ring type of fans, as the added resistance will increase the load on the motor, conceivably to a dangerous extent, if due allowance is not made for this factor (see also Chapter V., par. 12).

9. Velocity Considerations and Values.

In deciding on the velocities for the air movement to be adopted, both for the plenum and for the extraction parts of the system, some essential considerations are :—

(*a*) Limitations of economical speed of fan in the matter of power.

TABLE VIII.—**Suggested Velocities in the Air Circuit through Free Area.**

Part of Circuit.	Velocity (feet per minute).
1. Main intake for fresh air	500—900
2. Ducts from intake to tempering coils . .	800—1,000
3. Through tempering coils	800—1,500
4. Through washer	500
5. Through re-heater.	800—1,500
6. Main horizontal ducts from fan to branches .	1,000—1,500
7. Branches to vertical risers connecting to registers	800—1,000
8. Vertical risers (preferably velocity is 50 per cent. greater than (9) to secure a fairly equal discharge over whole area of register) . .	500—800
9. Through registers (depending on width of apartment and height of register above floor). Higher velocity should be used for wide rooms with registers 8 to 10 feet above floor level .	200—500
10. Extraction ducts	500—1,000

(*b*) Limitations of air velocities on account of noise and also loss of power in duct friction. (*Latter increases as the square of the speed.*)

(*c*) Limitations of air velocity through registers by reason of draughts (chiefly plenum) and noise.

(*d*) Desirability of as high a velocity as possible under the limitations above referred to, in order to reduce the size of duct to the minimum, and also to get a rapid conveyance of warm air with minimum heat losses. (Heat transmission through duct walls per foot super. per hour is practically constant, but at higher velocities the duct discharges more air at the same temperature, hence more heat units.)

Consider the maximum velocity in straight ducts when free from obstructions and bends to be 1,800 feet per minute, and through free area of registers to be 500 feet per minute, if silent operation is an essential feature.

10. Free Area.

By " Free Area " is meant the gross cross-sectional area of the duct, etc., through which the air passes, *less* the area of any obstruction, such as iron gratings, pipes, steel stanchions, etc.

Most careful attention must be paid to this definition, for otherwise failure is inevitable owing to the much higher values for resistances which will be introduced into the air circuit, which, since the fan pressure remains constant, will result in reduced quantity of air.

CHAPTER III

CONSIDERATIONS TO BE OBSERVED IN THE DESIGN OF THE AIR CIRCUIT

[A thorough understanding of the matter contained in this chapter is essential for the efficient design of the Air Circuit.]

1. Velocity Variations.

If an uniform stream of air moves along a duct in which the sectional area varies, the same weight of air must pass any two sections of the duct in the same interval of time, otherwise there would either be an obstruction or accumulation of a certain quantity of air in the space between the two sections. Thus, if at two points we have velocities v and v^1, and areas of a and a^1 we must have :—

$$av = a^1 v^1,$$

but from the well-known law of mechanics the energy stored in unit mass is—

$$\text{energy} = \tfrac{1}{2}mv^2,$$

or since mass (m) is equivalent to $\frac{w}{g}$ we have—

$$\text{energy} = \frac{wv^2}{2g},$$

where w equals the weight and g the acceleration due to gravity.

Hence there must be continuous changes in the kinetic (viz., motion) energy per pound of air, wherever a stream of it passes through a duct or channel of varying dimensions.

It can be shown that these changes are always accompanied by an increase or decrease in pressure and a reduction

(in either case) of total energy, the value of such reduction depending on the efficiency of the changes or conversion.

If a pipe through which air is moving with a certain velocity v, and containing therefore an amount of energy per pound weight—

$$\text{energy} = \frac{wv^2}{2g},$$

is allowed to discharge into the atmosphere when it is practically brought to rest, the whole of that energy is lost, as regards the air circuit with which we are concerned, and is converted into heat.

The case is quite otherwise, however, if the air (being confined to a duct) is allowed to come to rest gradually owing to the gradual enlargement of the duct, for then no local eddy currents are formed, which waste the energy of the moving air, but, instead, the gradual reduction of the velocity raises the pressure in front of the stream and communicates the increased pressure to the remainder of the air.

If the enlargement of the pipe is again reduced at the other end of the reservoir or tank in the same gradual way, the pressure accumulated by the gradual stopping of the stream is again utilised in starting a fresh stream without substantial loss. If the stream is allowed to pass into the expanded or reduced portion without careful design, *the loss of the entire energy of the moving stream must be counted upon.*

In the case of a bend, similar considerations hold if the air is driven up into a sharp angle, and a sudden change of direction is caused by a sharp bend; the whole of the kinetic (viz., motion) energy of the air in one direction is lost (as regards useful work on the air flow), such energy being transformed into heat energy instead of being utilised to do the work in hand. More energy is now required to start the air flowing in the new direction at right angles to the old direction, whereas had the change of direction been gradual the velocity of the air would not have been reduced and additional energy would not have been required.

2. Velocity and Pressure Head.

The flow of air through a duct or other closed channel is due to a difference in pressure existing between two points, such pressure usually being expressed in inches of water column in this country and ounces per square inch in the United States. If there were no frictional losses in the pipe, the total head or pressure set up by the fan or other apparatus would be available for producing flow, and the speed or velocity of that flow would be that which would finally result if a body under the action of gravity had fallen freely the distance measured by the head, or

$$v \text{ (in feet per second)} = \sqrt{2gh},$$

where h is the head or pressure in feet of *air* column.

Also $h = \frac{\text{H}d}{12\text{W}}$,

where H = head or pressure, expressed in inches of water column,

d = density of water,

W = weight of air in lbs. per cubic foot under standard conditions;

then substituting numerical values for the above expression we have

$$h = \frac{\text{H} \times 62{\cdot}37}{12 \times 0{\cdot}076} = 68\text{H},$$

substituting the value for h thus obtained we have—

$$v \text{ (feet per minute)} = 60\sqrt{2g\text{H}68},$$

whence

$$v \text{ (feet per minute)} = 3{,}950\sqrt{h},$$

and putting the static pressure H = 1 inch of water column from which

$$v = 3{,}950\sqrt{1}$$
$$v = 3{,}950 \text{ feet per minute for 1-inch pressure.}$$

The pressure head equivalent to the velocities can be obtained direct from Tables VII. or XXXVII.

Unfortunately it is not possible for either air or water (so far, these laws apply equally in both cases) to flow through a

pipe without some friction loss, however small, and the velocity of flow is, therefore, less than the formula would indicate. In other words, part of the total pressure exerted upon the air is employed in overcoming the resistance to flow, which is known as the static head, and part is utilised in the initial production and subsequent reproduction or maintenance of velocity at the sharp corners, etc., and is known as the velocity (dynamic) head.

It should be obvious, therefore, that, since it is the total head that is the only means of producing a flow of air in a duct or pipe, *the sum of the static and the velocity heads must equal the difference in the head* (i.e., *loss of draft pressure*) *between any two sections of a duct through which the air is flowing.*

With a constant total head, any increase in the length of pipe naturally increases the frictional loss (uses up part of static head available) and reduces the amount available for producing velocity (*i.e.*, output of air from the duct) accordingly. Similarly any abrupt change in sectional area or direction of the duct, right-angled elbow, etc., converts velocity head (kinetic energy) directly into heat, by reason of the eddies and whirls which are formed, thus causing still further reduction in the output of the duct.

The constant static head against which the fan delivers the air is often spoken of as the " Maintained Resistance."

CHAPTER IV

DUCTS AND REGISTERS

1. Provisions for Ducts on Plans.

Unless very great care is exercised in the design and construction of the ducts and registers, the ventilating system is doomed to failure. It is obviously more economical and less trouble to put down a much smaller plant, or entirely to omit to provide a ventilating plant, and hence *not* construct the building or audience hall in view, than to spend a relatively large sum of money on a first-class plant and then to reduce its output by 50 per cent. or so by reason of badly designed and constructed ducts, the plant and the system being blamed instead of the individual responsible for cramping the ducts.

In designing an audience hall or public building the following routine should be adopted if good results are to be obtained :—

(*a*) Select the site.

(*b*) Decide upon the general arrangement of the main apartments.

(*c*) Figure in tentatively the leading dimensions of these apartments and the seating accommodation.

(*d*) Estimate the quantity of air required adequately to ventilate each apartment.

(*e*) Recognise the *evil*, take it in hand, and figure in the dimensions of the main ducts and the proposed position of registers both for the plenum and extraction systems, so that these may receive due consideration in all future detail work, both constructional and decorative.

Nelson S. Thompson, of the Supervising Architects' Office, Treasury Department, which controls upwards of 650

Government buildings in the United States, in this connection advises :—

"The next step in the design of the mechanical equipment is the completion of the preliminary drawings . . . showing arrangements of the building.

"The mechanical draftsman will indicate on tracings all chases necessary for any part of the mechanical equipment, size, and location of all hot air flues and registers, cold air inlets, ventilators, vent ducts, flues and space required for boilers, fans, heaters, air washers and other machinery.

"These preliminary tracings are returned to the architectural draftsmen, *whose duty it is to arrange the building to receive the proper mechanical equipment.*"

A typical instance of this common-sense arrangement is useful further to illustrate the point, and Mr. Thompson continues :—

"In locating distributing cabinets (for electric light) in thin walls, care is taken to see whether a steel beam is located directly over the cabinet; and should this be the case, the structural engineer is requested to substitute two channels for the beam, and to set the backs of the channels $1\frac{1}{2}$ inches apart."

One cannot but remark what an improvement such a practice is to that unfortunately prevalent in this country, whereby the artist plans the building, including the decorative features, leaving the elevator, water, gas, sewerage, ventilating, fire equipment, vacuum cleaning, telephone, etc., engineers to do their best (or worst), fighting their every request for space, regardless of whether it may be reasonable or otherwise, and resulting in additional maintenance expenditure to the purchaser (which is the inevitable result of crowding all sorts of gear into the same chase), the inadequate provision of space for the efficient inspection and cleaning of elevator, etc., machinery, the wholesale covering up of inspection boxes (used for drawing in electric cables) by decorations, etc.

Instances continually arise where the area of a duct is reduced by 80 per cent. by a steel joist passing through it or the duct has four right-angled bends unexpectedly introduced into it to enable it to pass an obstruction, or possibly

it stops suddenly in a hollow ceiling or other space of at least twenty or thirty times the cross-sectional area, and then continues on the other side at least 50 feet away, the designer, constructor (or destructor!) being apparently totally ignorant of the fact that laws regulating velocity and pressure heads exist.

The duct work cannot be tucked into any old nook or corner or altered in area and direction to suit any and every tradesman engaged on the job, if it is to be successfully used for the purpose for which it was installed. *The air distribution system is part of a definite and highly intricate machine*, and if satisfactory results are to be obtained therefrom it must be treated with as much respect as a motor car, a dynamo, or a big printing machine. The law is inexorable.

No architect or engineer having selected a rolled steel joist on a factor of safety 2, would proceed to drill $\frac{3}{4}$ inch holes across both flanges and down the web; yet this is only the converse of providing a duct and choking it with obstructions, a common enough practice.

2. Materials for Construction of Air Ducts.

Air ducts are usually constructed by one of the following methods:—

(*a*) Bricks, preferably glazed or cement rendered, for the main vertical and horizontal ducts. For the vertical rises to registers brick ducts lined in galvanised iron as work proceeds are extensively used in the United States.

(*b*) Lath and plaster, with smooth side towards the air current.

(*c*) Galvanised iron for extraction ducts, and also for plenum work in factories, skating rinks, riding schools, expecially if heating is to be combined with ventilation on the hot-blast system.

(*d*) Eternite or other patent sheeting, or match-boarding fixed on timber or metal studding (difficult to make air-tight, mechanically weak for inspection

and cleaning purposes, and latter material is inflammable).

(*e*) Terra-cotta drain pipes of large diameter.

Whatever material is used—and in the opinion of the writer, for main ducts glazed bricks are preferable ; if these are too expensive, concrete cement rendered and white-washed for the smaller ducts, galvanised iron built into brickwork or not, as circumstances and finances permit—the two following rules should be rigidly observed :—

(*a*) *No square-cornered right-angled bends.*

(*b*) *No abrupt changes in shape or sectional area.*

In all cases access for adequately cleaning plenum ducts, preferably by means of hand brushes, or alternately by means of flue cleaning brushes, must be provided.

Attention should be paid to the subject of avoiding increased fire risk, by running ducts through fire-proof partitions and also preventing the undue transmission of sounds such as conversation, musical instruments, etc., also cooking and sanitary smells, from one apartment to the other by way of the ducts. This is a point where the long and specialised experiences of the ventilating engineer will be of inestimable value to him.

3. Pressure Losses in Air Ducts.

This subject was treated from a theoretical point of view in Chapter III. par. 2, but it is probably well to recapitulate here to a slight extent, as it is very important for successful design work.

Losses in duct systems or the air circuit are made up of two parts :—

(*a*) Velocity or dynamic losses.

(*b*) Static or friction losses.

Losses of velocity head are due either to changes in direction or velocity of the air, and are caused by the loss at entrance to duct, but principally by eddies and whirls set up in elbows and connections.

The loss at entrance is equivalent to the pressure head required to produce the velocity of the air in the duct, and

may vary from one to one and a half times the velocity head, depending upon the free area and whether the duct is connected directly to the fan outlet or whether through a plenum chamber. It may be expressed as a multiple of the pressure corresponding to the average velocity produced in the duct.

When the velocity in the duct is the same as at the fan outlet this may still be considered a loss, in view of the fact that, with a reduction of velocity through a gradually

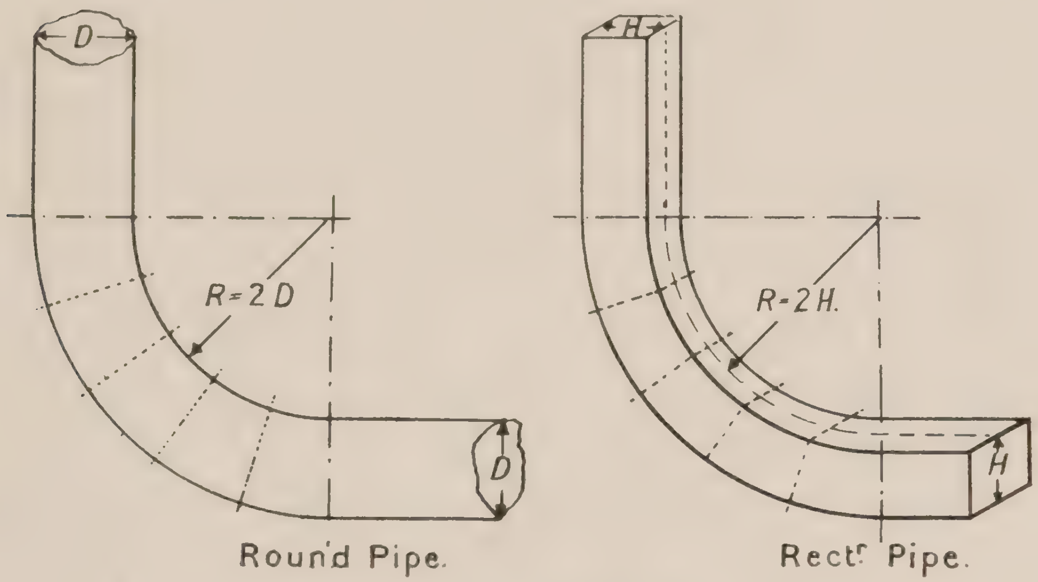


FIG. 7.—Resistance of 90° Elbows.

diverging outlet to a larger area, the difference between the velocity head at the fan and the velocity head in the pipe is largely utilised by conversion to static pressure.

The other chief source of *loss of velocity head* is in elbows, and *depends* directly *on the radius of curvature of the elbow*, and not on its size or the velocity of the air. For the purposes of calculation it is convenient to state this loss in the effect it would have in adding length of straight duct to the system—*i.e.*, equivalent lengths in diameters of round or widths (viz., longest side of rectangular section) of straight pipe. It will be seen that this is really finding a convenient method of stating loss of velocity head in terms of static loss, which simplifies figuring to a marked extent.

TABLE IX.—**Resistance of 90° Elbows** (Nelson S. Thompson).

Radius of Throat of Elbow in Diameters of Duct.	Number of Diameters of Straight Duct offering Equivalent Resistance.
— (square corners)	100·0
$\frac{1}{4}$	65·0
$\frac{1}{2}$	30·0
1	10·0
$1\frac{1}{2}$	6·0
$1\frac{3}{4}$	5·0
2 (maximum efficiency)	4·3

EXAMPLE 9.—For instance, a 90° elbow of 36-inch diameter pipe, having a radius of throat equal to 2 diameters—that is, 72 inches offers the same resistance to the flow of air as 4·3 diameters of straight pipe or 12·9 feet of straight pipe. By reference to the Friction Chart, Table XI. or Table XXXVI., the frictional resistance of the elbow can now be determined.

Bends in round or square duct should be turned on a true circle. In round ducts the crimped or one-piece elbow is not used, but they are made in sections.

Number of sections will be the *even* number (minimum of five) above the square root of the diameter of the duct, in inches.

EXAMPLE 10.—Thus the bend in a 60-inch diameter duct should be made up of

$$\sqrt{60} = 7{\cdot}74, \text{ viz., eight pieces.}$$

Branches, which should be of the same depth as the main pipe in rectangular or square ducts and be afterwards tapered as required, should leave the main duct at the side at any angle less than 45°, but preferably at 30°.

Outlets which discharge directly from the main or from branches, as is often the case in industrial buildings, should be made about two diameters in length.

In cases where the sectional area has to be increased or reduced, this, as has been shown, must be done without shock to the air current if loss of velocity head is to be avoided.

Efficient angles for the slope of the plates to effect the reduction can be obtained from Table X.

TABLE X.—**Loss of Pressure due to Converging or Diverging Nozzles or Cones** (W. H. Carrier).

Angle of Slope (in Degrees).	6	10	15	20	25	30
Efficiency of conversion (velocity pressure to static, or *vice versâ*) per cent.	70	56	42	28	14	—

For practical purposes make slope $1\frac{1}{2}$ inches (6°) to $2\frac{1}{2}$ inches (12°) to the foot, until the area is reduced or increased to size required.

The second source of pressure loss in the duct system or air circuit is due to the friction of the air rubbing against the sides of the duct. This loss will vary directly as the length of the pipe or as the square of the velocity, and inversely as the diameter of the pipe. As length is a fixed quantity for any system, the only factors subject to modification when working to obtain a definite friction (static) loss are the diameter of the duct and hence the velocity of the air, which are the factors which determine the relation between power cost for operating fan and cost of duct work.

It is the most convenient practice to determine a velocity which will give an assumed resistance considered suitable within the capacity of the fan (see Chapter II., par. 8).

The following formula indicates standard practice :—

$$P = \frac{1{\cdot}73\ K.S.V.^2}{A}.$$

Where P = Loss of pressure in inches of water;

K = A constant depending on the smoothness of duct surface, viz., 0·00012 for galvanised iron and 0·00022 for brick or concrete ducts.

V = Velocity of air, in feet per second.

A = Cross sectional area of duct, in square inches.

S = Rubbed surface in square feet.

For those who prefer it, this result can be obtained per 100 feet run of galvanised pipe for varying diameters of duct and air velocities from the chart of Table XI. or Table XXXVI.

Generally, the loss in static head in a length of duct equivalent to 60 diameters equals one velocity head.

4. Total Loss of Head.

The different sources of pressure loss for ducts only (exclusive of heaters, washers, etc.) have now been clearly shown, and therefore to ascertain the total loss of head in the duct system these separate losses must be collected as follows :—

(1) The pressure required to force the air through the plenum main intake and maintain the maximum velocity required in the air circuit. (It may be 1·0 to 1·5 times the head which is equivalent to the velocity.) It is only reckoned once unless there is a velocity loss due to a badly designed elbow or diverging or converging part of the duct, in which case deal with it as instructed in the next paragraph.

(2) Summarise all the velocity heads required in the ducts to produce increases of velocity due to variations in the section of the ducts. If all diverging and converging nozzles are well designed (slope $1\frac{1}{2}$ to $2\frac{1}{2}$ inches per foot), assume value of conversion at 50 per cent. in order to be on the high side. If in doubt, estimate on assumption that total kinetic energy of moving stream is dissipated—*i.e.*, that one velocity head is lost and hence that an additional one will be required in each case to restart the stream.

EXAMPLE 11.—Thus, if when the velocity is 1,750 feet per minute the duct has two right-angled bends and one badly designed enlargement, add the equivalent of one velocity head for each 1,750 feet per minute at 50° Fahr., viz., $0{\cdot}2 \times 3 = 0{\cdot}6$ inch water, in lieu of the increased length of duct method.

This energy loss can be visualised by reference to a roller skating rink on which a skater is seen gliding along one side of the rink. If he is a good skater, he will be able efficiently to change his direction and proceed along a line at 90° to his original direction without appreciable loss of speed, whereas a

An example will clearly illustrate the facility with which the friction can be determined from the chart, thus: Assume it is desired to discharge 40,000 cubic feet of air per minute through a duct 50 inches in diameter and 75 feet long. Find 40,000 cubic feet on right hand margin of the chart; follow horizontal line across to the left until it intersects with diagonal line marked 50 inches; now follow down vertical line marked to bottom of chart, where will be found the friction in inches of water, which in this case amounts to 0·31 inches per hundred feet of pipe. As the friction is in direct proportion to the length, then for 75 feet the friction will be: $\frac{0{\cdot}31 \text{ inches} \times 75 \text{ feet}}{100 \text{ feet}} = 0{\cdot}2325$ inches.

The chart can also be used for rectangular ducts by reducing the latter to round ducts having the same frictional surface in the walls of the pipe, thus:—

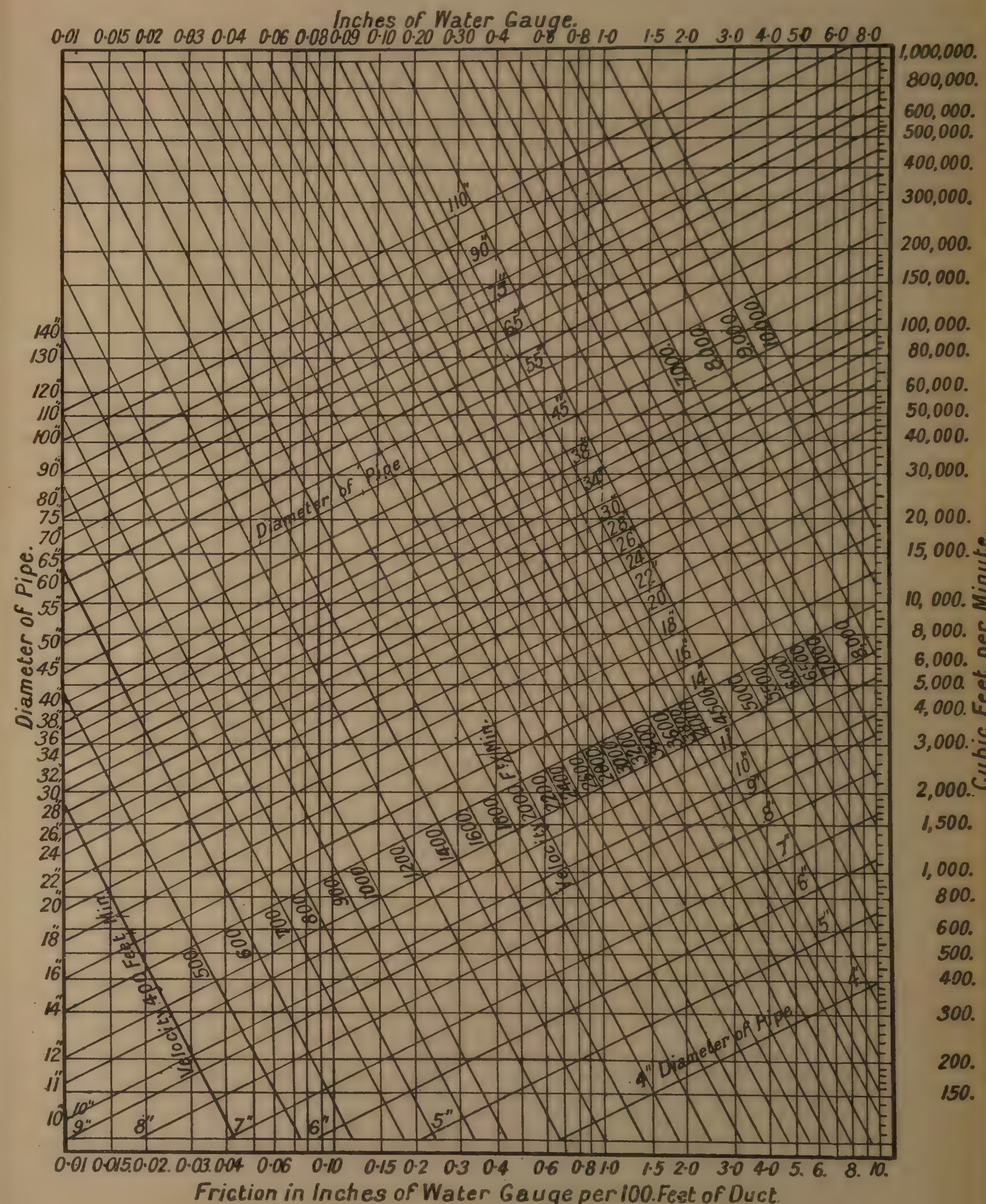
$$d = \frac{4\,a\,b}{2\,(a + b)}.$$

a = width of a rectangular pipe in inches.
b = depth of rectangular pipe in inches.
d = diameter of round pipe in inches, in which the air is flowing at the same velocity that will set up the same frictional resistance as the rectangular pipe.

The chart shows the friction considerably higher than it actually amounts to in a perfectly smooth round sheet iron pipe, to make allowances for dents, rivets and occasional roughness of surface.

The chart can be used for determining the diameter of a pipe of any volume at a given velocity; or the velocity can be determined for a given volume with a fixed diameter pipe; or the required diameter of a pipe can be determined, also the velocity, for a predetermined friction.

TABLE XI.—**Frictional Chart for Galvanised Iron Ducts.** (By the courtesy of The American Blower Co., U.S.A.)



beginner will run into the barrier, whereupon his speed vanishes and he loses the energy of motion. In order to get going again in the direction at 90° to his original path he has to use considerable energy to attain the speed at which he was previously skating along the straight. Here the loss in energy was obviously one velocity head.

(3) Having obtained the total equivalent length of duct (*i.e.*, actual length, plus allowances for elbows according to Table X.), the loss of head can be quickly ascertained from the chart of Table XI.

TABLE XII.—**Metal Gauges for Galvanised Iron Ducts.**

S.W.G. (English) . . .	26	24	22	20	18	16
Round diamr. (inches) . . .	15	30	42	54	72	above
Rectangular, longest side (inches) .	18	26	40	72	84	above

United States practice runs rather lighter than the figures given above, but, unless the job has to be cut in every detail, these weights should be adopted. Table XXXIX. gives weights for galvanised iron pipes and elbows.

For duct velocities see Table VIII., Chapter II.

5. Figuring Ducts of Varying Lengths and Duty for Equal Resistance.

All ducts connected in parallel on one fan system should be designed for approximately equal pressure drop so that they may pass the correct quantity of air with the minimum amount of permanent regulation.

There are many systems of figuring ducts for equal resistance—*i.e.*, long ducts, having higher resistance values than the average length or duty, are made of larger section to reduce the velocity and hence the static resistance. Conversely, ducts shorter or figured for lower duty than the

average have the air velocities increased slightly to increase the friction loss to the average value.

The following method, which is believed to be due to Thompson, appeals to the author as having the virtue of neatness, combined with accuracy and simple figuring :—

(1) Make main trunk at fan the same size as fan outlet (figured from maker's catalogue), or larger if desired to reduce velocity.

(2) Refer to Table XIII. and note how many 1-inch by 1-inch pipes are given as being equal in carrying capacity to this main duct.

(3) Divide this number of 1-inch pipes by the fan capacity, expressed in cubic feet per minute, which gives a fraction equivalent to the portion of 1-inch pipe to be allowed for each cubic foot of air.

(4) Multiply the amount of air to be delivered to each register by the equivalent distance of the said register from the fan. (Equivalent distance means total length of duct, after all additions for bends, contractions, etc., have been added.)

(5) Divide the total of these products by the fan output in cubic feet-minutes. This gives the average distance of all the registers from the fan.

(6) Multiply the amount of air to each register by the fraction obtained in operation 3, *i.e.*, the number of 1-inch pipes in the main trunk at fan. This will give the number of 1-inch pipes to be allowed for each register.

(7) *Corrections* :—

(*a*) For each 1 foot by which the equivalent distance (4) exceeds the average distance obtained in operation 5, one-third of 1 per cent. is to be added to the number of 1-inch pipes previously allowed for each register.

(*b*) If the equivalent distance is less than the average distance, the correction is to be deducted instead of added.

(8) Refer to Table XIII. and find the size of square pipe equivalent to the corrected number of 1-inch pipes for each flue. This is the size of the branch pipe at the base of the flue. Add all the branches back to the fan, or table, and if the figuring is correct it will add up the size of main trunk started with.

TABLE XIII.—**Equalisation Table of Ducts** (Thompson).

A. Diameter or side of square pipe in question (in inches).
B. Number of 1-inch square pipes equivalent to ditto.

A.	B.	A.	B.	A.	B.
1	1	20	1,788	39	9,498
2	5	21	2,020	40	10,119
3	15	22	2,270	42	11,432
4	32	23	2,537	44	12,482
5	60	24	2,821	46	14,351
6	88	25	3,125	48	15,963
7	129	26	3,446	50	17,678
8	181	27	3,788	52	19,499
9	243	28	4,148	54	21,428
10	316	29	4,528	56	23,468
11	401	30	4,929	58	25,620
12	498	31	5,351	60	27,886
13	609	32	5,793	62	30,268
14	733	33	6,256	64	32,768
15	871	34	6,741	66	35,388
16	1,024	35	7,247	68	38,131
17	1,191	36	7,776	70	40,996
18	1,374	37	8,327	72	43,988
19	1,573	38	8,901	74	47,106

This system would probably be of more use in hot-blast work when the distribution is by means of galvanised iron pipes, for the work spent on figuring would scarcely be repaid by results in public building work, where ducts are inevitably a compromise between structural details and a perfect system.

Example 12.—Fan with 54 × 54 inch outlet to handle, 28,000 cubic feet-minutes, feeding four registers.

Operation No. 1. 54 × 54 (inches).

" " 2. 21,000 (approx.).

" " 3. $\frac{21,000}{28,000} = 0{\cdot}75$.

" " 4 :—

Outlet No. (a)	Equivalent Distance. (b)	Cubic Feet per Minute. (c)	Equivalent Distance × Cubic Feet per Minute. (d)	Number of 1-inch Pipes. (Operation No. 6.) (e)
1	50	6,000	300,000	4,500
2	50	2,000	100,000	1,500
3	60	10,000	600,000	7,500
4	110	10,000	1,100,000	7,500
			2,100,000	

Operation No. 5. $\frac{2,100,000}{28,000} = 75$ feet (average distance).

" " 7 :—

Outlet No. (a)	Distance to be Corrected. (b)	Amount of Correction. (c)	Corrected Number of 1-inch Pipes. (d)	Size of Branches (inches). (e)
1	— 25	— 375	4,125	28 × 28
2	— 25	— 125	1,375	18 × 18
3	— 15	— 375	7,125	35 × 35
4	+ 35	+ 875	8,375	37 × 37

The main to carry branches Nos. 3 and 4 would be 8,375 + 7,125 = 15,500—viz., about 47 × 48 inches.

Rectangular pipes to be the same area when ratio of depth to breadth is not more than 3 to 1. If greater, correct as shown in Table XIV.

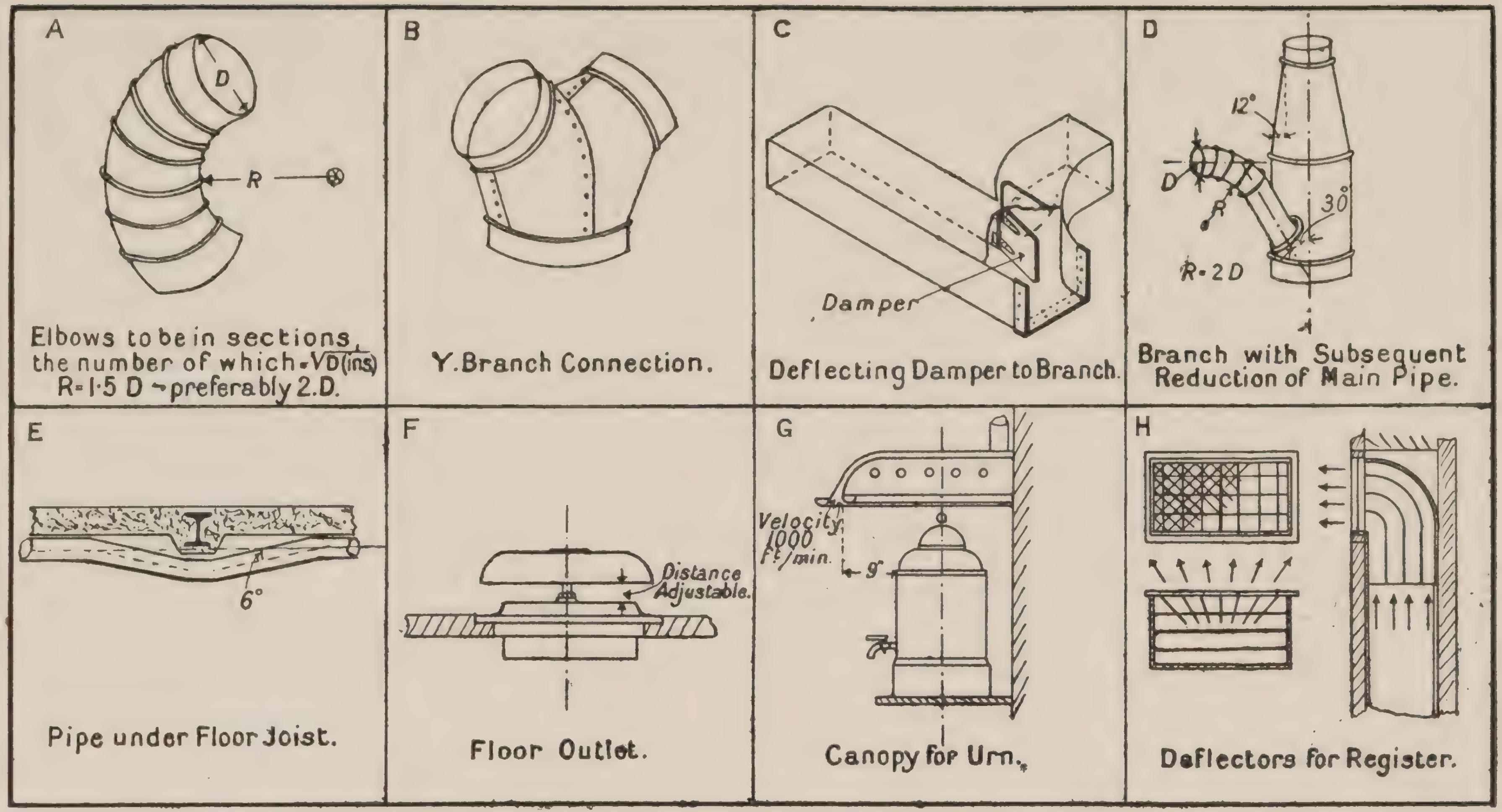


FIG. 8.—Standard Details of Sheet-Iron Duct-Work.

(A, B, C, D, and F, Sturtevont Engineering Co., Ltd.; G, Buffalo Forge Co.; H, American *Heating and Ventilating Magazine*).

TABLE XIV.—**Equalisation of Area of Round and Rectangular Ducts.**

Ratio of Length to Breadth (Rectangular)	4 to 1	6 to 1	10 to 1
Increase area by (per cent.) . . .	10	20	30

6. Registers.

As previously stated, the height for plenum registers above the floor level is 6 ft. 6 in. to 8 ft. 6 in. to the lower edge, the former being adopted in narrow apartments with lower discharge velocities and the latter for wider walls in which the higher velocities are used.

Baffle or deflecting plates materially assist in the equal distribution of air to all parts of the register and the control of the stream of air entering the apartment, thus avoiding any risk of complaints of draught from sensitive persons sitting near the register.

Screens of about No. 12 round wire formed into a 1-inch diamond mesh are useful to prevent paper and inflammable material being thrown into the duct through the cast-iron gratings used to cover the larger openings.

Excellent grids of standard sizes are listed by the National Radiator Co., Ltd., who state in their catalogue the "free or blast area" corresponding to the gross size—a valuable time-saving feature for busy designers.

CHAPTER V

FANS

1. Types.

The capacity of a fan is stated in cubic feet of air handled per minute, and the pressure set up in inches of water gauge.

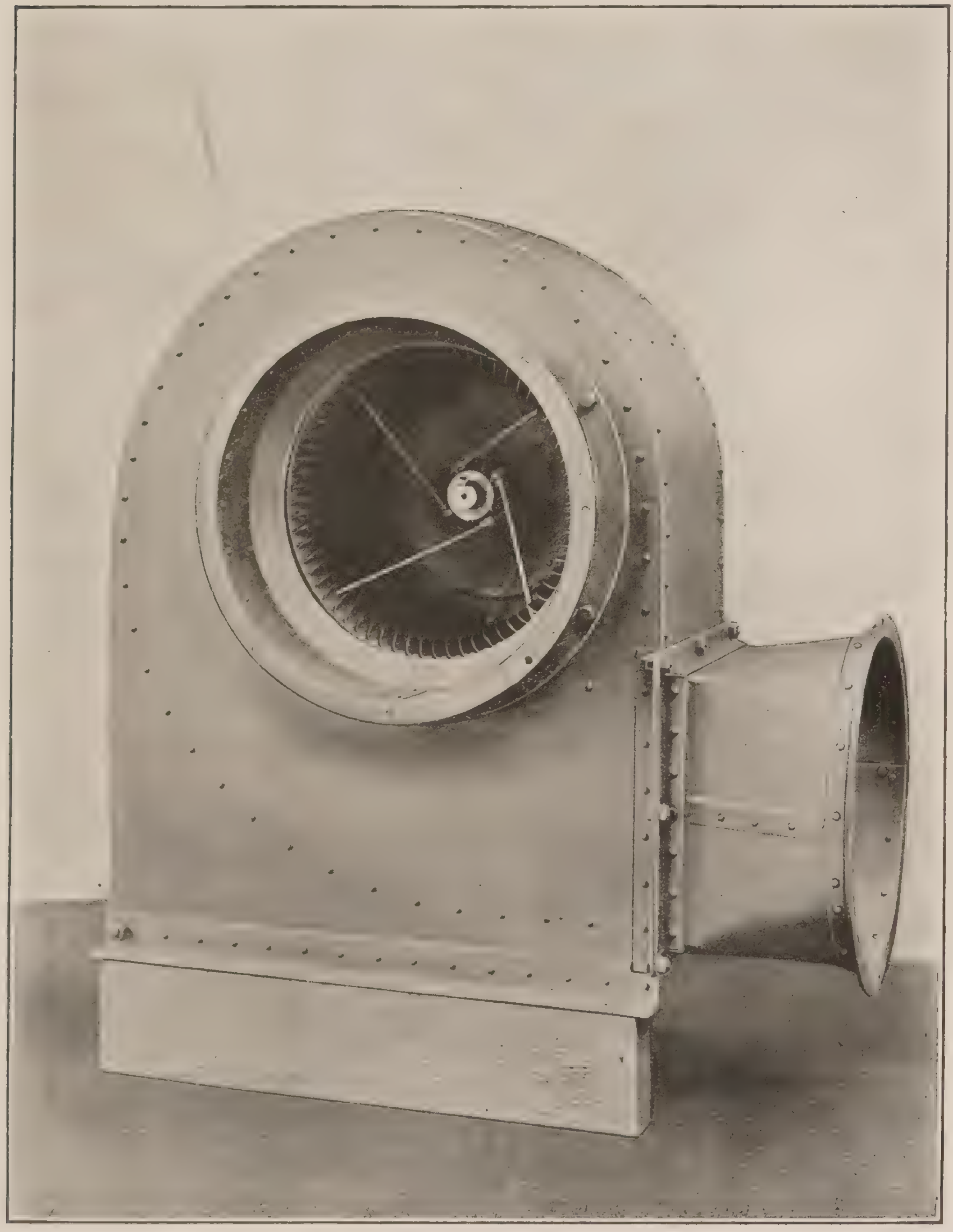


FIG. 9.—Single Inlet Multiblade Centrifugal Cased Fan.
(By the courtesy of Messrs. Davidson & Co., Ltd.)

There are two main types of fan in general use :—

A. **The Centrifugal Steel Cased Fan**, having not less than eight, and preferably many more, blades (*i.e.*, multiblade) for use where any appreciable resistance to the passage of the air is anticipated—*i.e.*, long ducts, heating coils, filters, etc., etc.

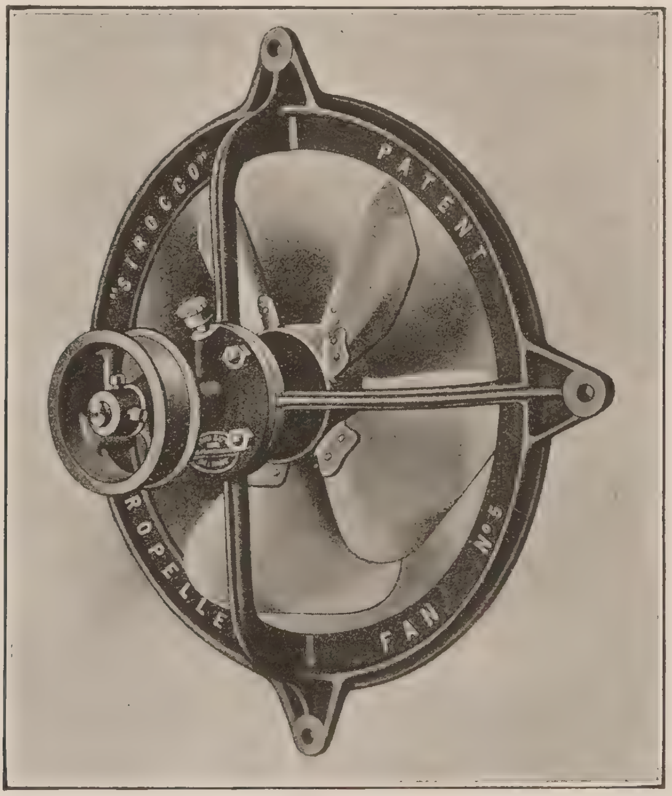


FIG. 10.—Ring Type Propeller Fan. (By the courtesy of Messrs. Davidson & Co., Ltd.)

B. **The Ring Type Fan**, which *is only suitable for handling volumes of air against negligible resistance*, *i.e.*, below 0·1 inch static water gauge, at an air velocity of the order of 500 feet per minute.

All ring type fans, whether of the box-blade, propeller, or porthole, etc., variety, should be arranged to discharge into or draw directly from the apartment without obstruction or against any strong aspirating tendencies. Numerous failures of ventilating installations can be cited which are entirely due to the use of ring type fans to deal with air

which it was desired to force through long ducts, and even, in addition, to draw it through cheese cloth filters and heaters.

2. Centrifugal Cased Fans.

For the purposes of plenum ventilation the centrifugal cased fan is selected, as it is required to deal with air at a medium resistance, the value of such resistance in a well-designed system being usually 1·00 to 1·50 inches static water gauge distributed over the air circuit generally, as shown in Table XV. :—

(N.B.—Hereafter water-gauge will be abbreviated to "W.G.")

TABLE XV.—**Loss of Pressure in the Plenum Air Circuit.**

	Inches W.G.
Intake ducts	0·10 to 0·20
Washer	0·25 to 0·35
Heater	0·20 to 0·30
Distributing ducts	0·25 to 0·40
Registers	0·20 to 0·25
Total static	1·00 to 1·50
Add for velocity head	0·30 to 0·60
Total head, *i.e.*, static plus velocity =	1·30 to 2·10

Makers put forward two principal types of the centrifugal cased fan, viz., the multiblade and the eight-bladed, usually known as the steel-plate fan. There are one or two other variations, notably the cup-shaped blade, for which its designers claim for it the special quality of "gripping the air and preventing it from sliding along to the side of the fan further from the inlet." Purchasers should exercise discretion and submit all claims to rigid tests of economical and noiseless running and workmanship.

The question of the number and shape of blades is largely a matter of individual choice, but the present writer inclines to the multiblade as made under the "Sirocco" patents

on the score of more silent operation, with fairly high efficiency in a more limited space. On account of its high speed also, it gives better pulley ratios in the larger sizes and is more suitable for direct coupling to electric motors in the smaller sizes.

There are, of course, various considerations in favour of forward, straight, backward curved, eight and multiblade, etc., types, but matter of this kind is rather outside the scope of the present work, as a certain amount of mathematics is involved, and readers interested are therefore referred to the authorities mentioned in the Foreword and also to Professor Carpenter's book, "Heating and Ventilating Buildings."

In the extraction system in cases where long and tortuous ducts are essential, similar pressure figures will be obtained to those given for the plenum system, if the heater and washer are omitted, and in this event a centrifugal cased fan must be used for mechanical extraction.

TABLE XVI.—**Loss of Pressure in the Extraction System.**

	Inches W.G.
Registers	0·01 to 0·25
Ducts to fan	0·15 to 0·30
Discharge	0·10 to 0·20
Total static	0·26 to 0·75
Add for velocity head	0·25 to 0·75
Total head, *i.e.*, static plus velocity =	0·51 to 1·50

In certain instances the figures for "ducts to fan" may be increased to 0·50 inch or even more on long runs; but these high values should be avoided wherever possible, as the interest on the increased capital cost of large ducts will usually more than counterbalance the decreased revenue cost of extracting air from high resistance ducts.

To ensure silent running and freedom from noise of "air rush" 1·25-inch and preferably 1·00-inch W.G. should

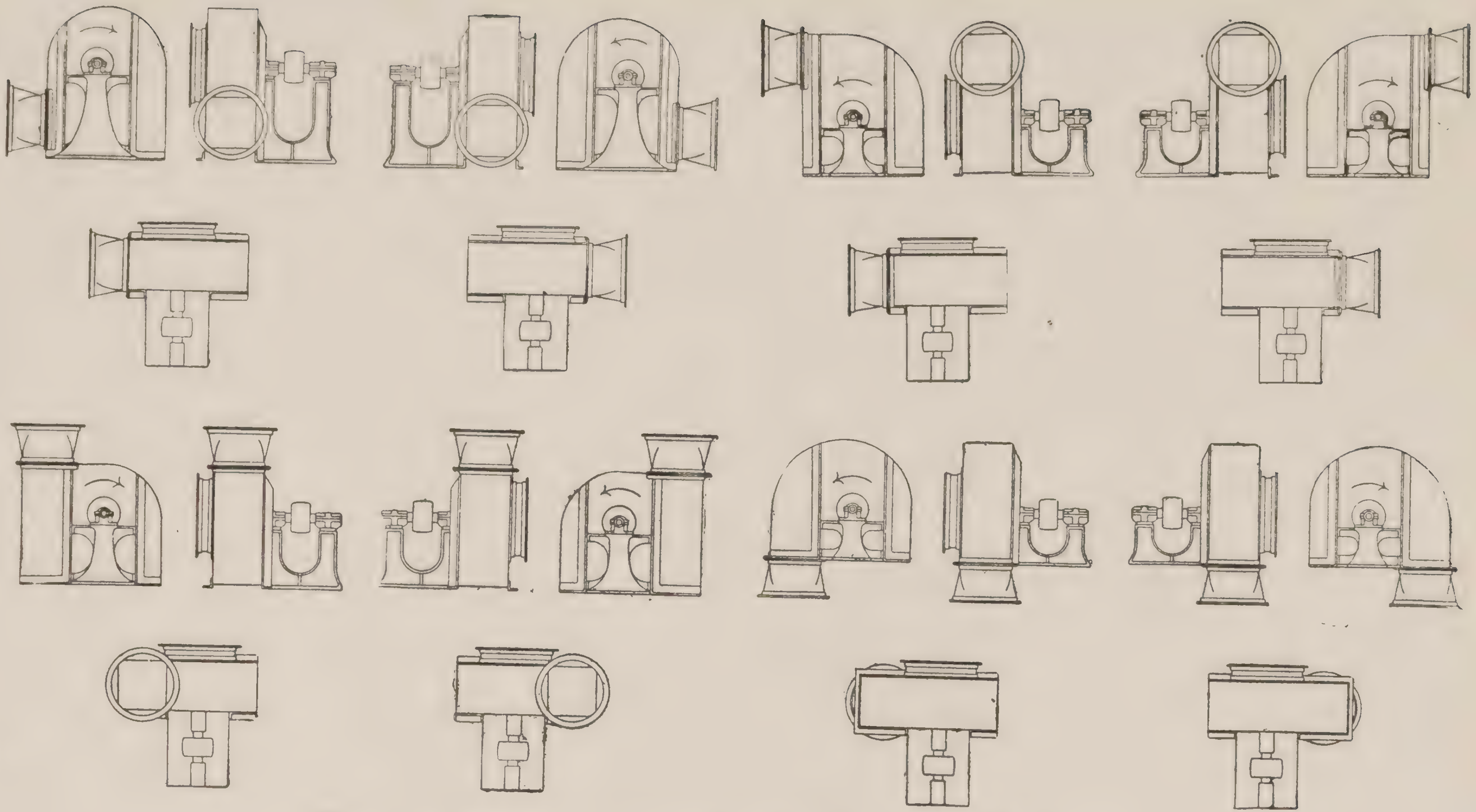


FIG. 11.—A Few Types of Centrifugal Cased Fans, (Davidson & Co., Ltd.).

be regarded as the maximum pressure at which the fan should work.

3. Designation of Centrifugal Cased Fans.

Centrifugal cased fans are made in a variety of forms in order to meet each particular requirement with a minimum of inconvenience.

Fans are designated right or left handed, according to whether the driving gear (pulley or motor) is on the observer's right or left hand side when looking into the discharge. "Top horizontal," "vertical up," "vertical down," etc., "discharge" are self explanatory.

Single inlet fans are usually preferred for plenum work or fresh air supply, as they are more easily arranged for a free, direct, and straight flow of air to the suction eye. Also, it is more simple to keep the bearings, belt, and motor away from the humid air current, thus avoiding rapid deterioration.

4. Double-width Centrifugal Cased Fans.

Centrifugal cased fans are also built of the double-inlet type, which are practically the same as two single-inlet fan wheels set back to back with the blade plate common to both sets of blades.

They have double the capacity of single-inlet fans of the same runner diameter, and are used where large outputs are required or head room is limited. Such fans are arranged with a bearing on each side of the casing, and are frequently used for mechanical extraction systems when appreciable duct resistance is anticipated.

5. Characteristics of Centrifugal Cased Fans.

For a given size of fan, air duct system, and air density (applicable to all centrifugal cased fans) :

(*a*) Capacity in cubic feet per minute varies directly as the speed.

(*b*) Velocity varies as the speed or capacity.

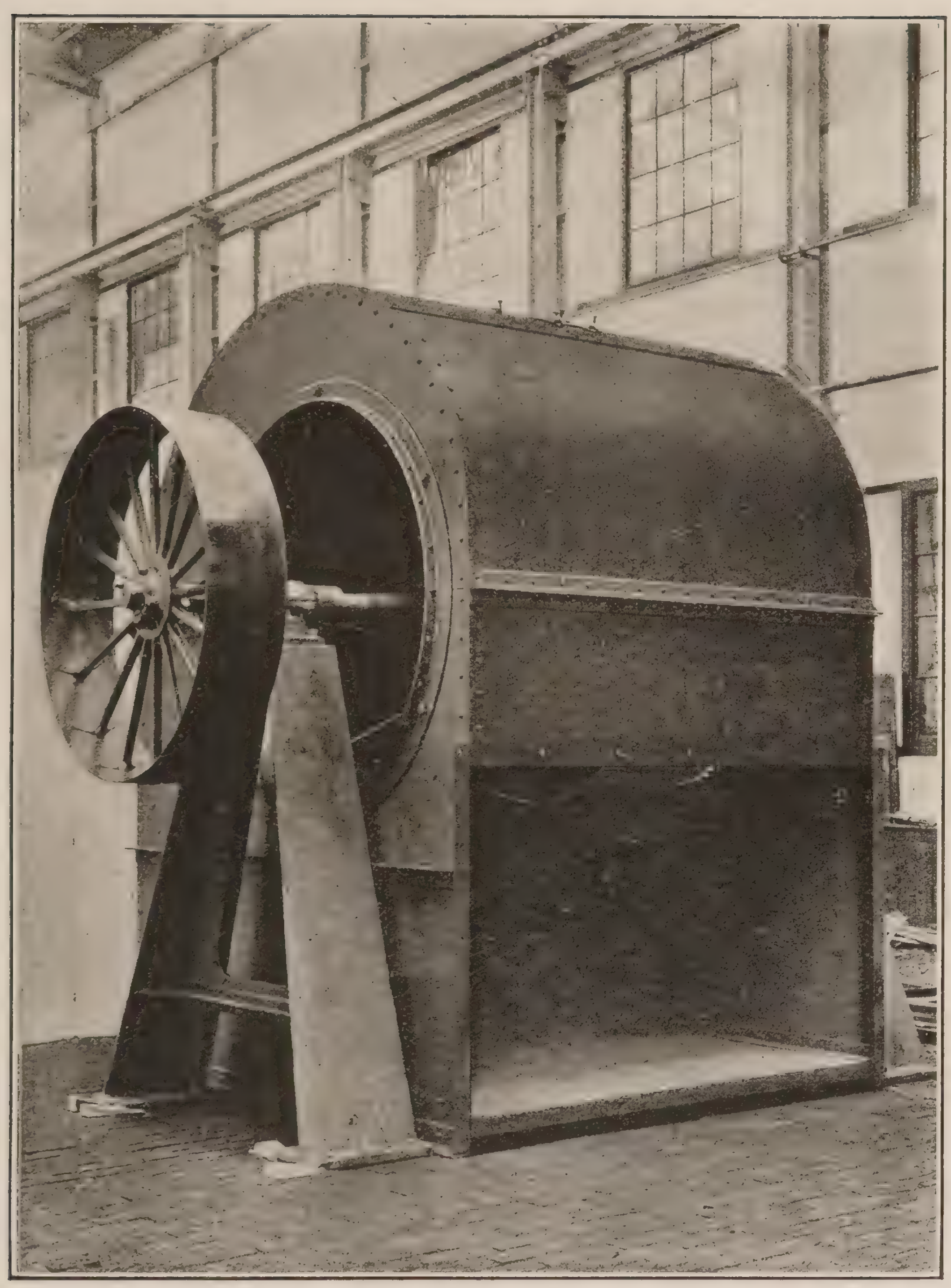


FIG. 12.—Double Inlet Multiblade Centrifugal Cased Fan, for a duty of $3\frac{1}{2}$ million cubic feet of air per hour against $\frac{1}{2}$-inch W.G. maintained resistance. (By the courtesy of Messrs. Davidson & Co., Ltd.)

(*c*) Pressure varies as the square of the speed.

(*d*) Speed and capacity vary as the square root of the pressure.

(*e*) Horse-power varies as the cube of the speed.

6. Pressure Ratings and Selection of Centrifugal Cased Fans.

Various makers adopt different systems of rating their fans, and this lack of uniformity is liable to lead to disappointing results if purchaser and contractor do not clearly understand each other when tenders are under discussion.

It is clear that work must be done in the air circuit for two purposes—

(*a*) To produce motion of the air, *i.e.*, the velocity at which the air must travel to attain its destination (after which the work is lost to the duct system, if not partially converted to pressure head previously).

(*b*) To maintain the velocity so produced, by overcoming the friction of the air rubbing on the sides of the ducts and tending to retard the motion of the air, *equally on the suction* and on the discharge sides.

Now, since we cannot produce velocity without the expenditure of energy on it, it is obvious that in considering the *total* efficiency of the fan we must include the pressure head due to velocity.

Further, since air will not remain in motion when opposed by resistance without the expenditure of further energy, we must add the head absorbed in friction (static), allowing thereon (Chapter IV., par. 3) for any equivalent head absorbed in reproducing the original velocities at badly designed elbows, converging and diverging connections, etc., so that we are forced to the following conclusions :—

(*a*) Total head to be developed by fan is the sum of velocity plus static head.

(*b*) Velocity or dynamic head is that portion of the total head used to produce air movement.

(*c*) Static head is that portion of the total head used to overcome friction, etc., losses.

When only the total efficiency of a fan is given, but the ratio of the static to the total pressure is known, the static efficiency may be calculated by multiplying the total efficiency by the ratio of the pressures.

Obviously, in public building work where velocities are comparatively low, it is high static efficiency that is required in order to overcome the duct, heater, washer, etc., friction. For each pressure there is a certain velocity through the fan outlet that will give maximum efficiency, and this holds true for all sizes having the same design of blade ; but of course it may be a matter of expediency to operate a fan at other than the most efficient point.

EXAMPLE 13.—Required to select a fan for a duty of 15,000 cubic feet per minute to be delivered against a maintained resistance (static) of 1 inch W.G.

Referring to the Buffalo Forge Co.'s Catalogue we obtain the following data.

TABLE XVII.—**Comparison of Performance of Three Fan Sizes for a given Duty.**

Fan. No.	Outlet Velocity.	Add for Total Pressure.	Capacity in cubic feet per minute.	R.P.M.	Mechanical H.P. required.	Air H.P. for Static Pressure.	Efficiency on Static Pressure (per cent.).
No. 8	1,600	0·16	14,930	298	4·51	2·34	51·8
,,	1,700	0·18	15,860	294	4·66	2·49	53·3
No. 7	2,000	0·25	14,290	330	4·08	2·24	54·9
,,	2,100	0·27	15,000	330	4·30	2·36	55·0
,,	2,200	0·30	15,720	329	4·56	2·47	54·1
No. 6	2,600	0·42	13,650	392	4·36	2·14	49·0
,,	2,800	0·48	14,700	400	5·00	2·31	46·3
,,	3,000	0·56	15,750	410	5·76	2·47	42·8

Where silence, low initial velocity and operating cost are desirable at the cost of capital expenditure No. 8 would be selected.

For moderate velocities and operating costs No. 7 is the fan ; whereas for high duct velocities and low capital cost with higher running cost No. 6 meets the requirements.

TABLE XVIII.—**Performance of Standard Single-Inlet Centrifugal Cased Fans.** (Davidson & Co.)

TABLE OF OUTPUTS.

Diam. of Discharge (in inches).	½-inch W.G.			¾-inch W.G.			1-inch W.G.			1½-inch W.G.			2-inch W.G.		
	R.P.M.	Cub. ft. per min.	B.H.P.	R.P.M.	Cub. ft. per min.	B.H.P.	R.P.M.	Cub. ft. per min.	B.H.P.	R.P.M.	Cub. ft. per min.	B.H.P.	R.P.M.	Cub. ft. per min.	B.H.P.
15	600	2,550	0·38	700	2,650	0·53	850	3,400	0·93	1,000	3,600	1·44	1,200	4,750	2·62
17½	550	3,600	0·60	650	4,120	0·91	750	4,800	1·42	900	5,500	2·33	1,000	5,400	2·88
20	450	4,350	0·64	550	5,020	1·07	650	6,250	1·84	800	7,650	3·38	900	8,050	4·43
25	350	6,300	0·87	450	8,600	1·91	500	8,900	2·45	600	10,400	4·19	700	11,400	6·08
30	300	10,100	1·49	375	12,800	2·60	425	13,500	4·2	520	16,500	8·00	600	19,000	10·8
35	250	12,400	1·70	300	14,800	2·98	350	16,700	4·45	450	23,000	9·73	520	26,000	16·0
40	225	17,300	2·51	275	21,100	4·55	325	26,200	7·62	385	29,000	12·0	450	33,600	18·5
45	200	22,800	3·36	250	28,800	6·37	300	36,400	11·5	350	38,300	16·2	400	43,000	23·6
50	175	25,300	3·47	225	35,200	7·79	250	36,800	10·1	300	40,000	16·0	350	47,900	25·5
55	160	31,400	5·00	200	40,100	8·56	225	44,500	12·2	275	51,200	20·5	325	63,700	36·3
60	150	40,800	6·02	175	43,600	8·79	225	64,800	20·4	260	67,300	32·0	300	78,000	43·7

Of course, the excessive portion of the velocity head of No. 7 can be more or less efficiently converted, depending on the angle of slope of the duct walls, into static pressure, but it is obviously useless to speed up at the fan in order to get a good fan efficiency if the value cannot be utilised in the air circuit. The point which concerns the engineer is the efficiency of the *complete* installation and not individual parts, except insomuch as they affect the whole.

English fan makers do not appear generally to have adopted the convenient methods of the American in listing fans showing the air velocity at outlet and the equivalent pressure to be added to the static value to obtain the total pressure, so that the Tenderer's Detail Sheet (Chapter XIV.) has been specially drafted to elicit this information.

It may be observed in studying fan performance tables or curves that there are two different outputs at the same static pressure for the same speed of fan, which would appear curious until it is noted that the total pressure is different, thus agreeing with the foregoing statement.

7. Ring Type Fans.

When the design of the air circuit is such that the fan works against negligible resistance or aspirating tendencies (below 0·125 inch W.G.), *i.e.*, with free intake and discharge, ring type fans are most effective, and are, of course, cheaper in first cost; but it should be clearly understood that *they are totally unsuitable for forcing into or extracting air from long or sub-divided flues*, and this type of fan should therefore only be used in places where it can discharge into or exhaust the air direct from the ventilated apartment without obstruction. For silent running the maximum peripheral speed should be of the order of 4,000 feet per minute.

Although not so efficient as centrifugal cased fans when operating against resistance, yet because of low first cost, large air capacity, and simplicity of erection, ring type fans are largely used for roof extraction, ventilating engine-rooms,

kitchens, ship's cabins, etc. They can be obtained direct coupled to electric motors, which is very convenient for

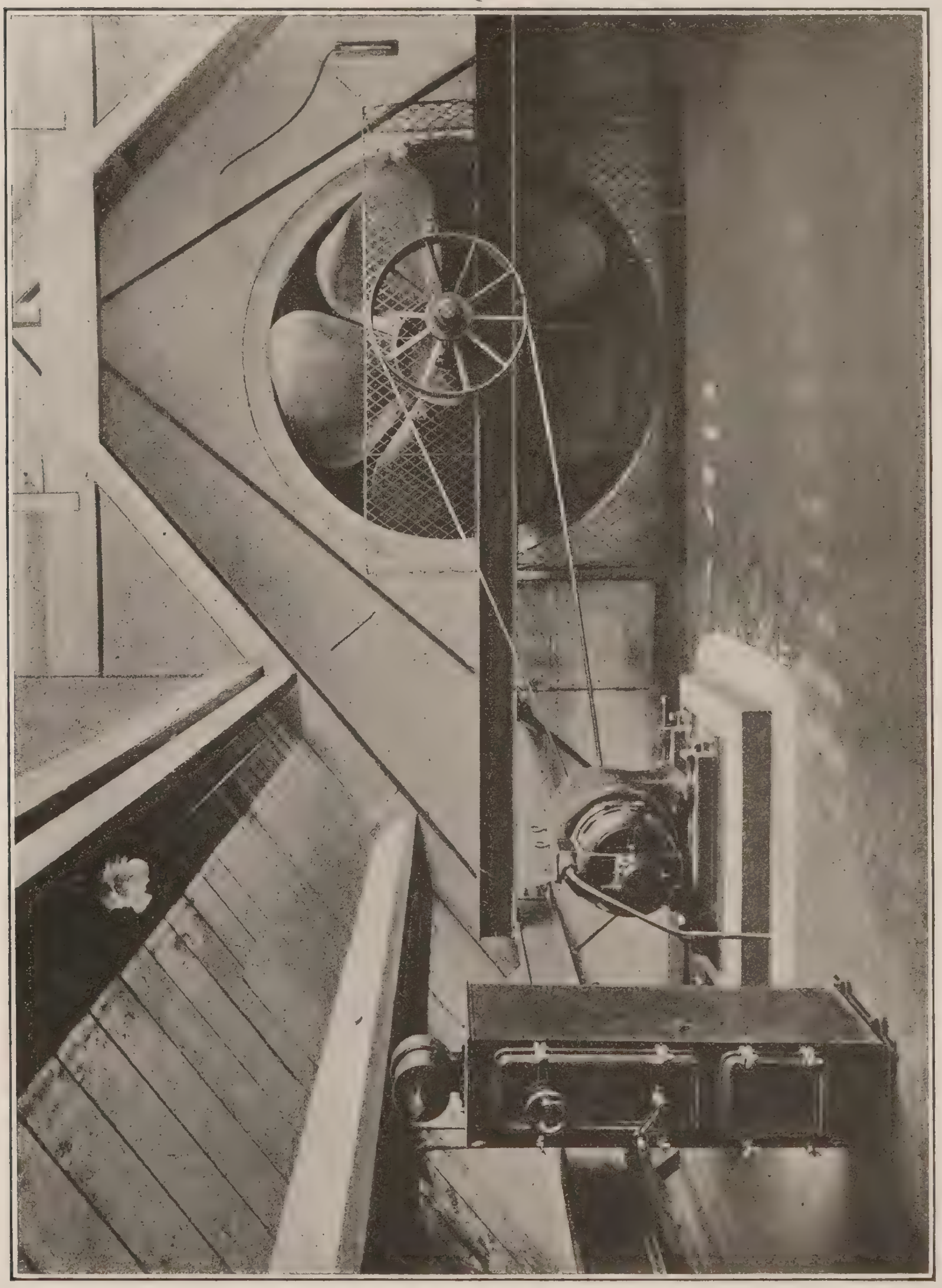


FIG. 13.—Ring Type Propeller Fan (arranged for belt drive), for a duty of $3\frac{1}{2}$ million cubic feet of air per hour against free intake and discharge.

the smaller sizes, or arranged for belt drive in the larger ones.

8. Frames for Ring Type Fans.

Various makers' designs of ring type fans differ, some wheels being suitable for propelling air with equal efficiency, either inwardly or outwardly, according to the direction of rotation, but others require slight rearrangement. Attention must therefore be paid to this point in the specification when ordering.

The frame ring and frequently the arms are all cast in one piece, three or four lugs, through which bolts pass, for attachment of the frame to the wall, partition or window being provided. This design is usual for fan wheels up to 60 inches diameter. Above this size it is customary to provide two bearings either grease or oil ring lubricated, one at each side of propeller wheel, carried on wall brackets and steel joists to reduce vibration and prevent undue stress on the one bearing and on the wall.

9. Capacities and Powers of Ring Type Fans.

TABLE XIX.—**Performance of Standard Ring Type Fans.**

(Slow speed for silent running.)

Fan Wheel Diameter (inches)	Revolutions per Minute	Peripheral Speed of Runner (feet per minute.)	Volume of Air in cubic feet per minute at *free Intake and Discharge.*	Brake Horse-power at Fan Pulley.
14	1,000	3,650	1,000	0·10
18	800	3,750	1,800	0·12
24	600	3,780	3,000	0·17
30	500	3,920	5,000	0·25
36	450	4,250	8,000	0·50
42	400	4,400	11,000	0·62
48	350	4,400	14,000	0·75
54	300	4,230	18,000	1·0
60	280	4,400	24,000	1·2
66	250	4,320	28,000	1·5

A large margin of safety should always be allowed when deciding the capacity of ring type fans, as their operation is materially affected by many apparently trifling things.

10. Characteristics of Ring Type Fans.

Variations in the output for any particular fan wheel diameter can easily be obtained by raising or lowering the speed, where—

(*a*) For a constant diameter, angle of blades, and position of fan the volume varies directly as the speed.

(*b*) Horse-power varies as the speed cubed.

(*c*) Box-bladed and screw-type wheels have a capacity 25 to 30 per cent. greater than a disk or flat-bladed wheel of the same diameter.

(*d*) The air velocity is usually 16·0 to 21 per cent. of the peripheral speed.

Readers desiring to study the subject further should turn up Mr. W. G. Walker's Paper, read before the Institution of Mechanical Engineers, 1897.

11. Fixings for Ring Type Fans.

As previously noted, wheels up to 60 inches diameter are usually fitted with cast-iron ring frames for carrying the single bearing. In this case a brick ring or wood frame is formed to the dimensions and the design shown on the makers' blue prints, and the ring is bolted directly to the wall or partition, which should be finished dead true to avoid straining the frame when fixing bolts are tightened. Short lengths of gas barrel built into the wall to receive the fixing bolts when brick seatings are used facilitate erection.

All bolt hole centres and the other dimensions required can be obtained from the makers and special scale drawings supplied after the fan is ordered, care being taken to see that drawings are marked "FINAL," and not "PRELIMINARY," otherwise dimensions taken off the actual fan as delivered may differ from those shown on the drawings.

The suction side of the fan must not be unduly choked, a clear space equal to half the diameter of the wheel being left all round the ring to obtain maximum efficiency, as *air is drawn in a direction not only parallel to the shaft, but also at right angles to it.* Especially is this the case with ring type fans fitted with box-blade wheels.

12. Variable Duty for Centrifugal Cased and Ring Types of Fan.

The characteristics of the centrifugal cased fan are such that if the discharge be stopped by closing a damper the load falls rapidly, the wheel continuing to rotate and to increase the static pressure slightly above the normal by reason of the centrifugal force which the rotation of the air round the axis generates, but as no air is passing through the fan the power consumption is materially reduced, and the rotor with the contained air acts as a sort of fluid flywheel. When the damper is opened slightly some air passes ; hence the power consumption increases. Although this method of obtaining reduced output is convenient, it is not so economical or so quiet as that to be obtained by varying the speed, which can easily be done by installing a variable speed shunt-controlled motor on direct current electricity supply systems. On an alternating supply the damper device is practically the only arrangement available for obtaining reduced outputs unless cone pulleys be used. Dampers to these should be fixed on the discharge and not on the suction side of the fan, if on suction side what is known as surging takes place.

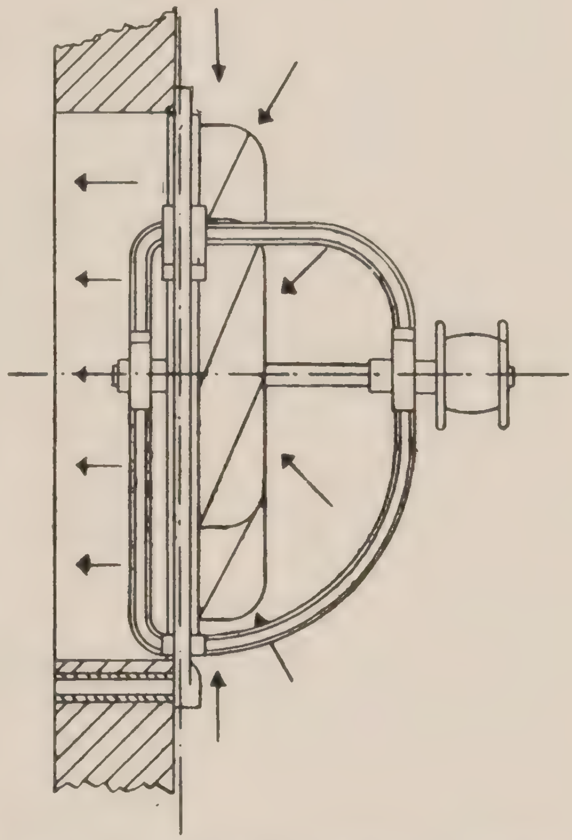


Fig. 14.—Air Flow through Box Blade Fan.

The ring type fan (of which the wheel is some portion of a screw propeller), contrary to the centrifugal cased fan, exerts a direct thrust on the air, and therefore increased

resistance to the flow of air results in an increased power consumption, so rapid, in fact, that it is perfectly possible for such a fan to take nearly three times as much power with the damper closed as with the damper fully open. This effect is not so noticeable with some of the curved box or propeller bladed fan wheels, which have to some extent the characteristics of centrifugal cased fans. In any case, regulation is not efficiently carried out by direct damper control, and if a variable speed motor cannot be used, then by-pass dampers which allow a part of the air to circulate round the fan instead of drawing all from the ventilated apartment must be installed if variable duty is required occasionally. Another and possibly better alternative in a situation of this kind is to provide two fan systems, one designed for say one-third and the other two-thirds of the full duty. If this arrangement is adopted care must be exercised in selecting the positions for the fans to avoid the risk of them pulling against each other, automatic shutters being provided for each, thus closing the outlet when the fan is not in use.

Fifty to one hundred per cent. speed regulation where obtainable is desirable for full control, the maximum speed of the fan corresponding to a discharge about 10 to 15 per cent. in excess of the estimated quantity of air required.

13. Precautions to be Adopted to ensure Silent Running.

The term " noiseless running " as applied to machinery is purely one of degree and largely a question of mechanical isolation.

Naturally, prevention is better than cure in fan engineering as in many other things, and for this reason it is desirable to avoid—

(*a*) Running fans at high speeds to get high pressures in order to overcome the resistance of badly designed ducts. 1·25 inches—preferably 1·00—W.G. static pressure should be considered, the maximum to be set up by the fan.

(*b*) Avoid high velocity of air and rapid changes in the direction of air ducts. Consider 1,500 feet per minute—preferably 1,000—to be the maximum, that is, except at the actual intake and discharge of the fan, where the velocity values may be in excess of this figure—possibly 2,000—2,500 feet per minute.

(*c*) Fix fans on solid foundations and avoid any large expanse of hollow concrete floor which could possibly act as a sounding " board."

(*d*) Avoid the vicinity of steel joists forming part of the framework of the building. These are frequently found to be efficient sound carriers.

(*e*) Adopt endless belt drive above 3 B.H.P., so that if the motor is at all noisy, being away from the duct, the air currents will not carry the noise forward.

(*f*) Stop long or echoing ducts a foot or two short of the fan, and make the gap good with painted canvas matchboarding, etc.

(*g*) Bed fans and motors on cork and felt pads, and isolate from the main structure by slipping sleeves over the holding-down bolts and washers of the same material under the steel washers, fixed under the nuts, care being taken that no part of the foundation bolts are in contact with the fan frame or base plate.

14. Power required to Drive Fan.

Work is done whenever motion is the direct result of a force being exerted against a resistance, the British unit being the weight of 1 lb. lifted through 1 foot (*i.e.*, 1 foot-lb).

The amount of work done is independent of time, but *power*, which *is the rate of doing work*, is inversely proportional to the time (*i.e.*, the faster the work is done, the greater the force required).

The British unit or horse-power has been fixed at 33,000 foot-lbs. per minute.

Now, since air has weight (0·075 lb. per cubic foot at 60° Fahr.) and is moved against a definite resistance (measured for convenience in inches of water column) in

given time, power is exerted a value for which is found as follows :—

$$\frac{\text{Cubic feet of air per minute} \times \text{pressure in inches (W.G.)} \times 5{\cdot}2}{33{,}000} = \text{H.P.}$$

or $0{\cdot}000157 \times$ cubic feet of air per minute $\times$ pressure in inches (W.G.) = H.P.

N.B.—To obtain total efficiency W.G. must include static and velocity head.

One cubic foot of water, $12 \times 12 \times 12$ inches deep, weighs 62·4 lbs. Hence a $12 \times 12 \times 1$ inch slab weighs $62{\cdot}4 \div 12$, viz., 5·2 lbs., the figure given above.

The mechanical efficiency of fans varies considerably, but is usually of the order of 45 to 70 per cent., depending on size and design.

CHAPTER VI

AIR WASHING AND HUMIDIFYING PLANT (FILTERS)

1. Air Conditioning (Humidification and Cooling).

On account of the dust and soot introduced by a ventilating system some form of air washer or air filter is essential where cleanliness is of importance and the spray type has superseded the old and insanitary cloth, coke, and other types of filter on account of its increased efficiency. Ability to regulate the quantity of moisture contained in the treated air and also to lower the temperature 7° to 10° Fahr., if required, are also valuable features of this type of plant, which is rapidly becoming standard practice for first-class ventilating work.

In view of the unique experience which Mr. N. S. Thompson has had, and of which he has made the fullest possible use in dealing with the United States Government buildings, the writer ventures again to quote this engineer's views, which he has found fully confirmed in practice :—

"In all buildings equipped with a plenum fan the practice is to install an air washer, as such a device is looked upon as the *most essential* feature of a successful ventilating apparatus. The old plants were a failure on account of the inability of the dry air filter to cleanse properly."

One of the chief objections to steam and hot-blast heating in the past has been due to the failure of hygienists to appreciate the importance of the humidity of conditioned air.

It has already been pointed out (Chapter I., par. 9) that the amount of moisture which air can hold increases rapidly with the temperature, so that if, during winter weather, air at a temperature of 30° Fahr. and 70° relative humidity (1·4 grains of moisture per cubic foot)—a very

usual figure—is passed through a heater in which the temperature is raised to 65° Fahr., the relative humidity

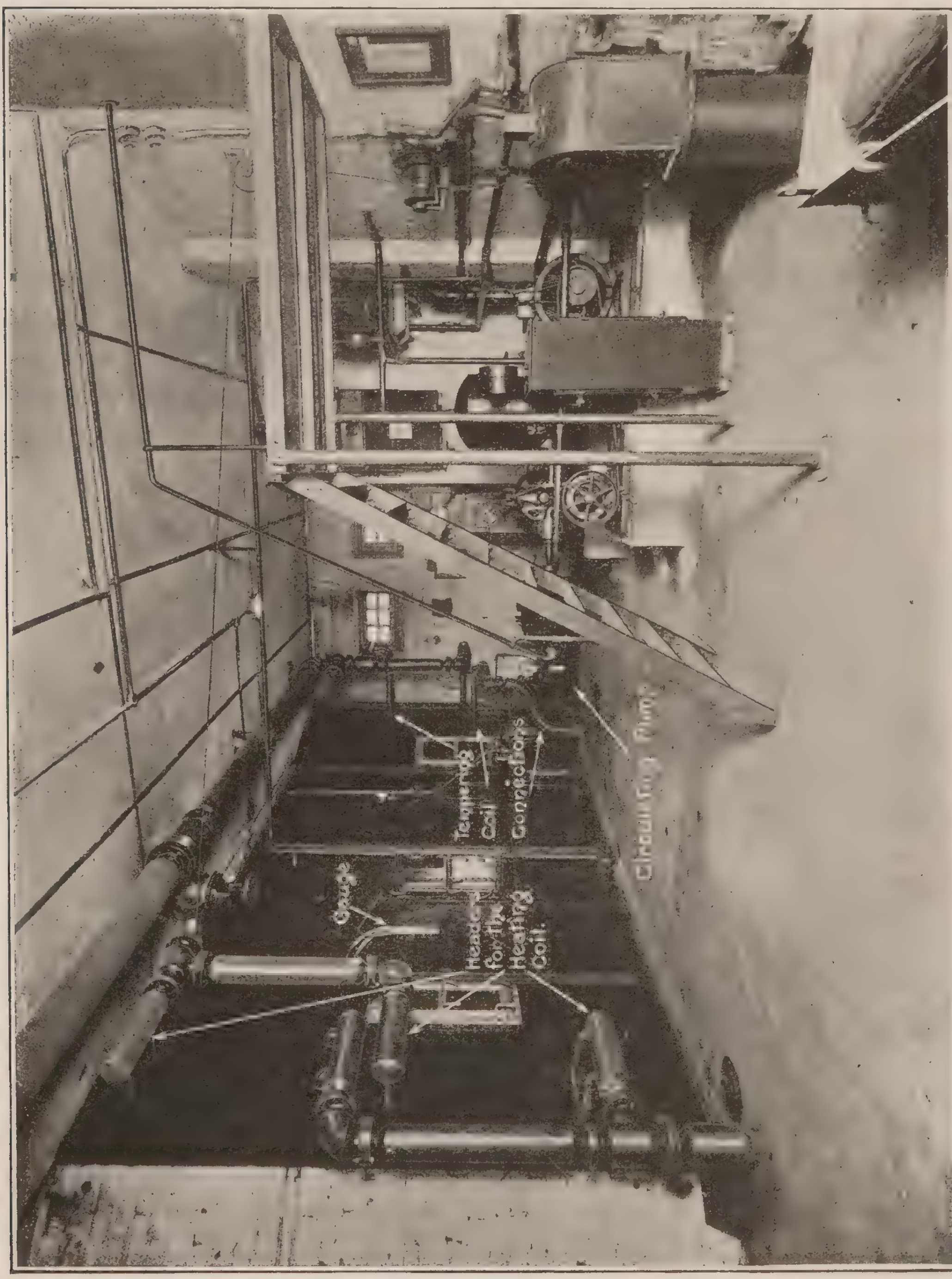


FIG. 15.—Wet Air Filter (general view), for a duty of $3\frac{1}{2}$ million cubic feet of air per hour.

will, unless moisture be added, fall to 20 per cent. of saturation, and complaints of dry and parched throats, etc., will inevitably result. A modern air washer would, under these conditions, add approximately 1·5 grains of water vapour

per cubic foot of air to be conditioned, thus maintaining the relative humidity at 43 per cent. at the higher temperature equivalent to 50 per cent. at 60° Fahr., at the temperature of the apartment, and by this means cause the air to be delivered in a state in which it is pleasant and comfortable to breathe.

The most desirable degree of relative humidity to maintain in public buildings, audience halls, etc., is from 45 to 60 per cent. at a room temperature of 60° Fahr. (43 per cent. at 65° Fahr.), corresponding to an average saturation temperature or dew point of 41° Fahr. and a total moisture contents of 2·9 grains per cubic foot of air, which is equivalent to a wet bulb depression of 9½° Fahr. This value will still cause a certain amount of condensation on the windows in very cold weather, and if such condensation for any reason has to be avoided, then the relative humidity at 60° Fahr. must temporarily be reduced approximately to 35 per cent., corresponding to a dew point of 30° Fahr., a total moisture contents of 2 grains per cubic foot of air, and a wet bulb depression of 13° Fahr.

In Chapter I., par. 10, it was shown that cooling can be effected by the evaporation of water. In warm weather a shade temperature of 85° Fahr. and a relative humidity of 40 per cent., corresponding to a moisture contents of 5 grains per cubic foot, is frequently experienced. If the moisture contents be increased to 6 grains per cubic foot the temperature will be reduced 8½, or, say, 9° Fahr.—viz., to 76° Fahr.—the corresponding relative humidity to this weight of water vapour and air temperature being 62 per cent., which is quite comfortable. A greater cooling effect would be obtained by the continuous use of fresh cold water, or in a greater degree by the introduction of refrigerating coils in the water tank. From practical experience, however, a drop of 10° to 15° Fahr. between the actual temperature of the apartment and outside air is found to be the maximum, greater differences causing complaints of cold even on the hottest days, by reason of the sudden change from one temperature to the other (American practice).

This interesting question of cooling audience halls, public buildings, etc., to any appreciable degree is one that has received comparatively little attention in this country up to the present, but, given an artesian well yielding a plentiful supply of relatively cold water, 50° to 55° Fahr., reasonably near the surface, it would be a simple matter not only to avoid increase of temperature of the ventilated apartment due to bodily heat, but also to effect a reasonable degree of cooling on a commercial scale. For details of cooling by mechanical refrigeration see Chapter I., par. 10.

2. Constructional Details of Wet-Air Filters.

A modern well-designed wet-air filter for use in connection with the ventilation of public buildings will carry out the following work :—

(*a*) Remove 98 per cent. of the solid material carried by the entering air.

(*b*) Increase the relative humidity to 85 per cent. of saturation point at the temperature of the spray water.

(*c*) Remove fog to a very marked degree.

(*d*) Eliminate any odours which are soluble in water.

(*e*) Remove all trace of free moisture from the conditioned air (a valuable feature in the ventilation and cooling of the electrical machinery).

(*f*) When using cold water (for cooling purposes), reduce the outgoing difference of air and water temperatures to less than 25 per cent. of the difference in the incoming temperature.

Modern air washers consist of seven principal parts :—

(*a*) The Spray Chamber.

(*b*) The Scrubbing Plates.

(*c*) The Eliminating Plates.

(*d*) The Settling Tank (containing the Primary Filter).

(*e*) The Secondary Filter.

(*f*) Circulating Pump (usually electrically driven).

(*g*) Heating Surface in the Settling Tank.

The leading dimensions of one of a well-known make of air washers are as follows :—

TABLE XX.—**Constructional Details of Wet-Air Filters** (Davidson & Co.).

Air Capacity.	Weight.	Casing Dimensions. Length.		Width.		Height overall.		Pump. Capacity.	Horse-power.
cub.ft.-min.	tons.	ft.	in.	ft.	in.	ft.	in.	gallons. per min.	
4,000	1	7	0	3	9	3	8	20	$1\frac{1}{2}$
6,000	$1\frac{1}{4}$	7	0	4	3	4	4	30	$1\frac{3}{4}$
8,000	$1\frac{1}{3}$	7	0	4	3	5	5	40	2
10,000	$1\frac{1}{2}$	7	0	5	3	5	5	50	$2\frac{1}{2}$
15,000	$1\frac{3}{4}$	7	0	6	3	6	5	75	$3\frac{3}{4}$
20,000	$2\frac{1}{4}$	7	0	6	9	7	5	100	$4\frac{1}{2}$
30,000	3	7	0	8	10	8	6	150	6
40,000	$3\frac{3}{4}$	7	0	10	4	9	6	200	8
50,000	$4\frac{1}{2}$	7	0	10	4	11	6	250	10
60,000	$5\frac{1}{4}$	7	0	12	4	11	6	300	12

(It will be noted that the length is constant for all sizes, the width and height being the only variable dimensions, in order to obtain constant air velocity through the washer, approx. 500 feet per minute.)

The Spray Chamber occupies that part of the washer nearest the main intake and contains a system of galvanised water piping, to which are attached, uniformally spaced over the cross-sectional area of the washer, spray nozzles, having large orifices and discharging in the direction of air flow. They produce a cloud of water in a very finely atomised condition, so that the whole chamber is filled with a thick mist, and the air in passing through is thoroughly mixed with it.

The Scrubbing Plates, which are placed at the end of the spray chamber, are fixed at an angle to the normal direction of air flow, and are kept constantly flooded with water,

thus effectually removing the solid and dissolving the soluble impurities. This leaves the air in a saturated condition, *i.e.*, containing a large percentage of free moisture or water in the liquid form.

The Eliminating Plates, situated immediately behind the scrubbing plates, are also fixed at an angle to the normal direction of air flow, and are, in addition, frequently provided with lipped edges, so that the air in its passage through both the scrubber and eliminating plates is forced against them and on the latter, which are not flooded, all remaining particles of free moisture (drops of water) are deposited, the air being discharged in the condition described. It is essential, in selecting any particular design of washer, to see that the plates are well designed, to avoid baffling the air current and setting up material resistance to the flow, an average value for this being 0·25 inch with 0·375 inch as a maximum—preferably the former figure.

> EXAMPLE 14.—$\frac{1}{8}$ inch additional pressure on a 60,000 cubic feet plant running 3,000 hours per year requires at 1*d*. per horse-power hour
>
> $$\frac{60{,}000 \times 0{\cdot}125 \times 5{\cdot}12 \times 3{,}000 \times 1d.}{33{,}000 \times 240},$$
>
> viz., £14 15*s*., assuming 100 per cent. for efficiency, or, say, £22 on a practical figure.

The Settling Tank, usually constructed of galvanised plate iron, stiffened by an angle framework of the same metal, acts as the base of the filter and contains the water supply used by the sprays. A ball-cock valve connected to the supply main provides the make-up water (required by reason of evaporation) automatically maintaining the proper level. Rapid filling, overflow, and waste connections are also fitted.

As will be seen from Fig. 16, the tank is divided into two parts by a screen of fine wire mesh, through which the water has to pass on its way to the spray system *viâ* the secondary filter and the pump. This primary filter removes all comparatively large débris—straw, cinders, feathers, rag, etc.—

which may be carried in through the main intake, from the water. Watertight glass doors and electric lights are usually supplied to enable the apparatus to be adequately inspected

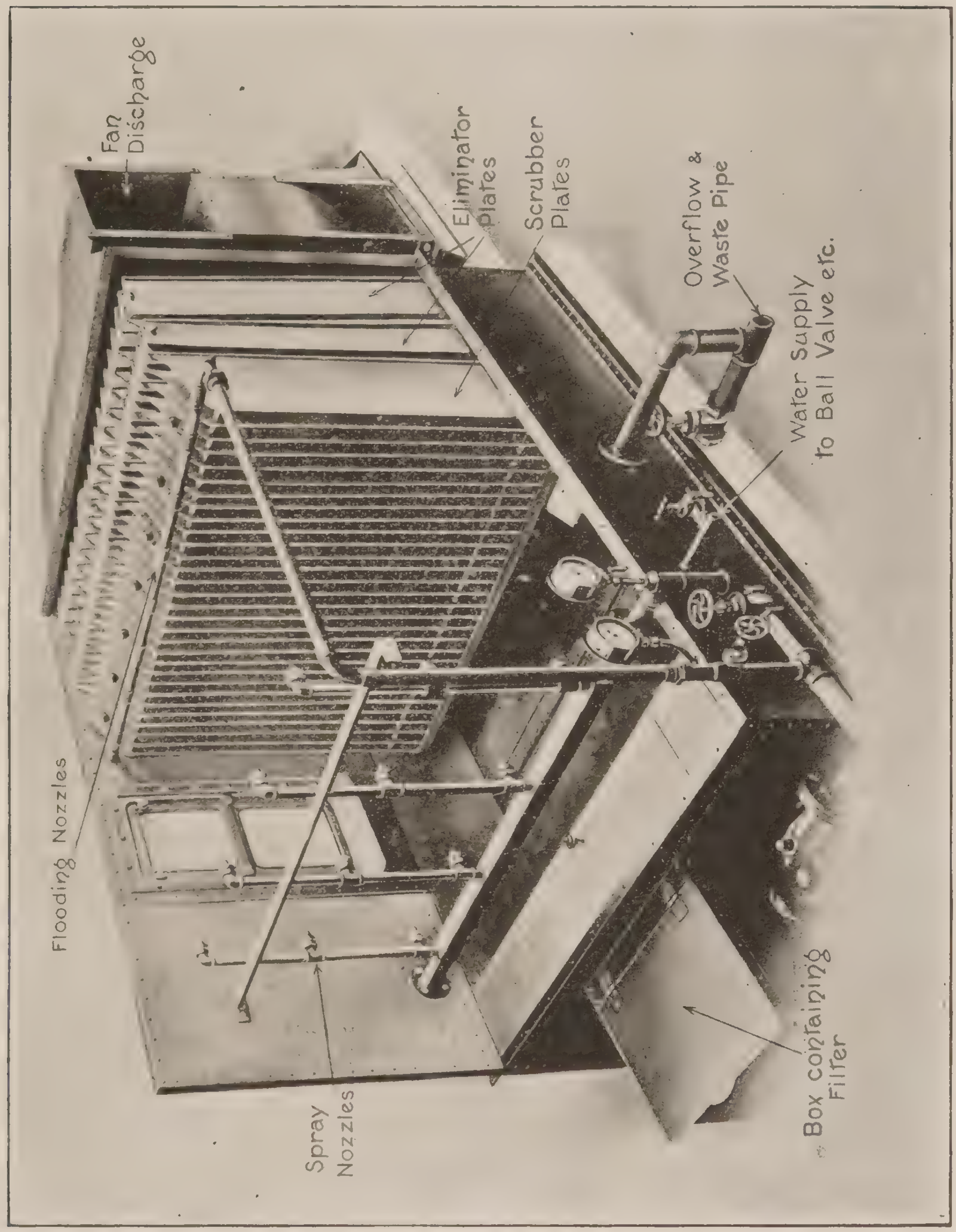


FIG. 16.—Wet Air Filter (detailed view). (By the courtesy of Messrs. Davidson & Co., Ltd., Belfast.)

when running, and for access for periodical cleaning, which may be required daily or weekly, depending on the cleanliness or otherwise of the air to be treated.

The secondary filter varies in design, but is inserted in the

pipe line between the gauze screen (referred to in the preceding paragraph) and the pump in order to remove effectually all solid matter remaining in the water, which would be liable to clog the sprays. In this connection it should be noted that the relative size of gauze mesh in the filter system and the smallest water way in the spray nozzle is an important factor in continuous working. The twin pattern filter, as fitted by Messrs. Davidson to the "Sirocco" washer, is an ingenious device which enables the strainers to be removed one at a time and cleaned while the pump is running.

The Circulating Pump, usually of the centrifugal type, direct coupled to an electric motor, constantly re-circulates the water used by the sprays, the pressure adopted on the delivery being 25 lbs. per square inch for the atomisers and 8 to 10 lbs. per square inch on the scrubbing nozzles. Since for efficient filtering the spray chambers must be filled by the fine mist of water, it is not possible to reduce the output of the pump when the fan is running at a slow speed for half, three-quarters, etc., duty.

Whenever possible the pump and motor should be so fixed that the top of the suction flange on the pump casting is below the bottom of the tank in which case the pump will always be flooded when the tank contains water, and by this means the necessity for fitting foot valves and priming cocks is avoided.

3. Design Details.

In considering various tenders for wet-air filters attention should be paid to the under-mentioned points :—

(*a*) Air must be brought into most intimate contact with a very finely divided spray.

(*b*) Large water-ways required in spray nozzles to avoid stoppage ; insist on $\frac{3}{16}$ inch diameter as minimum.

(*c*) Nozzles discharge water in direction of air flow, thus avoiding excessive humidification.

(*d*) Adequate filtering of water when it is to be re-circulated.

(*e*) Uniform distribution of air over the whole cross-sectional area of the washer. This is most efficiently effected by arranging a straight entry of the air into the washer and by providing a perforated metal plate (say 1-inch diameter holes) at the entrance.

(*f*) Saturated air, after leaving the spray chamber, should be broken into very narrow layers to be brought into repeated contact with the wetted surfaces of the scrubber.

(*g*) Lips, or some efficient substitute, to be provided on the elimination plates to remove completely all free moisture.

(*h*) It will be found that the humidifying effect depends on the water pressure maintained on the spray nozzles. This is useful on warm or humid days, as when less moisture has to be added the pumping pressure is reduced by means of the check valves.

(*i*) For some types of water, containing a percentage of organic acids, copper plates and fittings are necessary. The Water Supply Department are usually glad to advise on this matter.

4. Degree of Humidification.

As will be readily understood, from Chapter I. par. 9 and par. 1 of the present chapter, the regulation of the degrees of humidification is largely a matter of temperature control, and this is dealt with fully in the next chapter.

CHAPTER VII

HEATERS

1. Systems of Heating Apartments.

There are two methods in general use of warming an apartment, *i.e.* :—

(*a*) *Direct Heating*, in which the heat transmission losses through walls, windows, etc., are provided for by means of radiators fixed in the rooms to be heated, the air for ventilation purposes being warmed only to a sufficient degree to prevent complaints of draughts by reason of convection currents, and to avoid undue cooling of the ventilated apartments.

(*b*) *Indirect Heating* by hot air (or hot blast, as it is termed in the United States), may be adopted, in which case the air to be supplied must be heated to a sufficiently high temperature (maximum 130° Fahr.) to provide the heat transmission losses through wall, glass, etc., in addition to that required for humidification and tempering of the air. Air when treated in this way frequently acquires a "burnt flavour," and it is also stated that it has an injurious effect on the nervous system.

The former system (direct heating) is the one generally adopted in United States and British Government Buildings, the Metropolitan Asylums Board, banks, insurance offices, etc., and recently installed in the new Regent Palace Hotel, the Royal Automobile Club, the Midland Adelphi Hotel, Liverpool, etc., and is advocated by the present writer after careful comparative tests of both systems. It is interesting to note in this connection that direct heating surface has, in many instances, been after-

wards added to systems originally designed for indirect heating only, although, of course, for factories, cinema studios, skating rinks, and buildings of similar type hot-blast heating is frequently preferable by reason of the possibilities of warming large areas when slight draughts are not a serious source of complaint, considered in conjunction with the absence of radiators, pipes, etc., which would otherwise form obstructions to the movement of travelling cranes, erection of scenery, shafting, machine tools, the transport of materials in course of manufacture, etc.

In favour of the direct system, it may be said that any accident to the fan, motor, or plenum battery will not cripple the entire heating system, and the apartment may still be usable. Again, the large motors driving the plenum fans, the circulating pump on the air washer, etc., are only required when the apartment is actually in use, and not for a five or six-hour preliminary warming up, a financial consideration when electricity is the motive power.

2. Heat Units required for Air Conditioning.

As stated in Chapter I., pars. 6 and 8, the limits of temperature through which the air has to be handled are 30° Fahr. intake (minimum) to approximately 65° Fahr. discharge to ducts for ventilating installations working in conjunction with direct heating systems in the British Isles, thus allowing for 5° drop in transit through the ducts, which is an average figure. It was also shown that within these limits, and without serious error, as compared with the increased facility for figuring, it could be considered that 1 *B.Th.U. will raise the temperature of* 53 *cubic feet of air by* 1° *Fahr.* under standard conditions of pressure.

As regards humidification, with the type of washer described (*i.e.*, not specially built for humidification) the air will probably leave the eliminator within 15 per cent. of complete saturation at that temperature (*i.e.*, 85 per cent. relative humidity), so that in order to obtain a final condi-

tion of 65° Fahr. and 43 per cent. (or 60° Fahr. and 50 per cent.) relative humidity, which corresponds to a moisture contents of 2·9 grains of water vapour per cubic foot of air, the spray chamber must be maintained at a temperature of 45° Fahr., corresponding to a dew point of 40° Fahr. Or conversely, if 2·9 grains represents 85 per cent., then 3·4 grains represents 100 per cent., and the temperature must be such that, if working at 100 per cent. efficiency—viz., if the leaving air was saturated—it would contain 3·4 grains, yet at 85 per cent. efficiency it only contains 2·9 grains per cubic foot.

In order to obtain maximum values, assume the intake air to be at a temperature of 30° Fahr. and 70 per cent. relative humidity, corresponding to a moisture contents of approximately 1·4 grains of water per cubic foot of air, so that the weight of water to be added is 2·9 grains (final) less 1·4 grains (initial), viz., 1·5 grains of water per cubic foot of air treated. The figures are necessarily based on the assumption of existing washer efficiencies, but these may be improved as their use becomes more general.

It can be shown experimentally that the temperature of the air will drop approximately $8\frac{1}{2}$° Fahr. for each grain of moisture absorbed per cubic foot of air treated (this is slightly higher than the theoretical figure); hence the drop in temperature to be anticipated by the absorption of 1·5 grains is approximately 13° Fahr. per cubic foot, and the *tempering coils must be designed to raise the temperature of the intake air from* 30° *Fahr. to* 45° *Fahr. required for humidification plus* 13° *for cooling effect*, viz., 58° *Fahr.*—*i.e.*, an increase of 58° less 30°, viz., 28° *Fahr.* In some cases, the heating surface required for humidification is placed in the settling tank of the air washer, thus controlling the spray temperature, but this course is not recommended.

The *re-heating coils* will be smaller, as they *are only required for the duty of raising the temperature of the air* from that prevailing in the spray chamber to the final temperature, viz., *from* 45° *Fahr. to* 65° *Fahr.*, or *an increase of* 20° *Fahr.*

TABLE XXI.—**Heat Units Required for Conditioning Air** (Tempering 30° to 65° Fahr. and Evaporation of 1·5 Grains of Water per Cubic Foot : Air in Cubic Feet : Heat in B.Th.U. per hour).

Air per Minute.	Air per Hour.	Preliminary Heating.	For Evaporation.	Total for Tempering Coils.	Re-heating Coils.	Total Heat of Battery.	Boiler Horse-power (30,000 B Th.U. = 1 B.H.P.).
1.000	60,000	17,000	14,700	31,700	22,600	54,300	1·80
2.500	150,000	42,500	36,800	79,300	56,500	135,800	4·52
5.000	300,000	85,000	73,500	158.500	113,000	271,500	9·05
10.000	600,000	170,000	147,000	317,000	226,000	543,000	18·1
20.000	1,200,000	340,000	294,000	634,000	453,000	1,087,000	36·2
30,000	1,800,000	510,000	440,000	950,000	680,000	1,630,000	54·3
50,000	3,000,000	850,000	735,000	1,585,000	1,130,000	2,715,000	90·5
75,000	4,500,000	1,270,000	1,100,000	2,370,000	1,700,000	4,070,000	135·

3. Heating Medium to be Employed.

The usual methods of transferring the heat from the fuel *viâ* the furnace to the air to be conditioned are by :—

(*a*) Hot Water.
(*b*) Steam.
(*c*) Hot-Air Furnace.

Of these the two former are in general use, but the latter finds little favour in modern ventilating installations (although largely used for the purpose of tea and vegetable drying plants) owing to the difficulty of convenient application and the "burnt flavour" usually associated with air treated in this way.

Hot water, which *must* include for mechanical (accelerated or pumped) circulation of the water in order to keep the size of the heater and pipe connections within practical limits, has many advocates, the present writer having used it extensively, but low-pressure steam is almost equally popular for plenum battery work.

Steam systems admit of simple, reliable, and rapid humidity control, and, in view of the fact that the plenum heater frequently forms nine-tenths of the total load on the heating boiler, it is obvious that any irregularity in its working due to the failure of a fan motor, etc., will result in violent fluctuations of temperature on the direct heating system. Hence it is a distinct advantage to keep the two systems entirely separate (especially when batteries of boilers are installed), in which case the large steam boiler need only be fired about an hour before the fan is required, whereas the hot-water boiler may be run within a reasonable margin of its full rating continuously for the direct heating surface radiators, depending on weather conditions, and in this way reduced fuel consumption will result.

Low-pressure steam heating (5 to 10 lbs. gauge pressure) for plenum batteries is favoured on the grounds of—

(*a*) Possibilities of rapid variations of temperature,

and hence humidity, due to the comparatively small heat capacity of the medium actually in the heater.

(*b*) Practical impossibility of failure through freezing in severe weather, due to absence of liquid in well-drained systems.

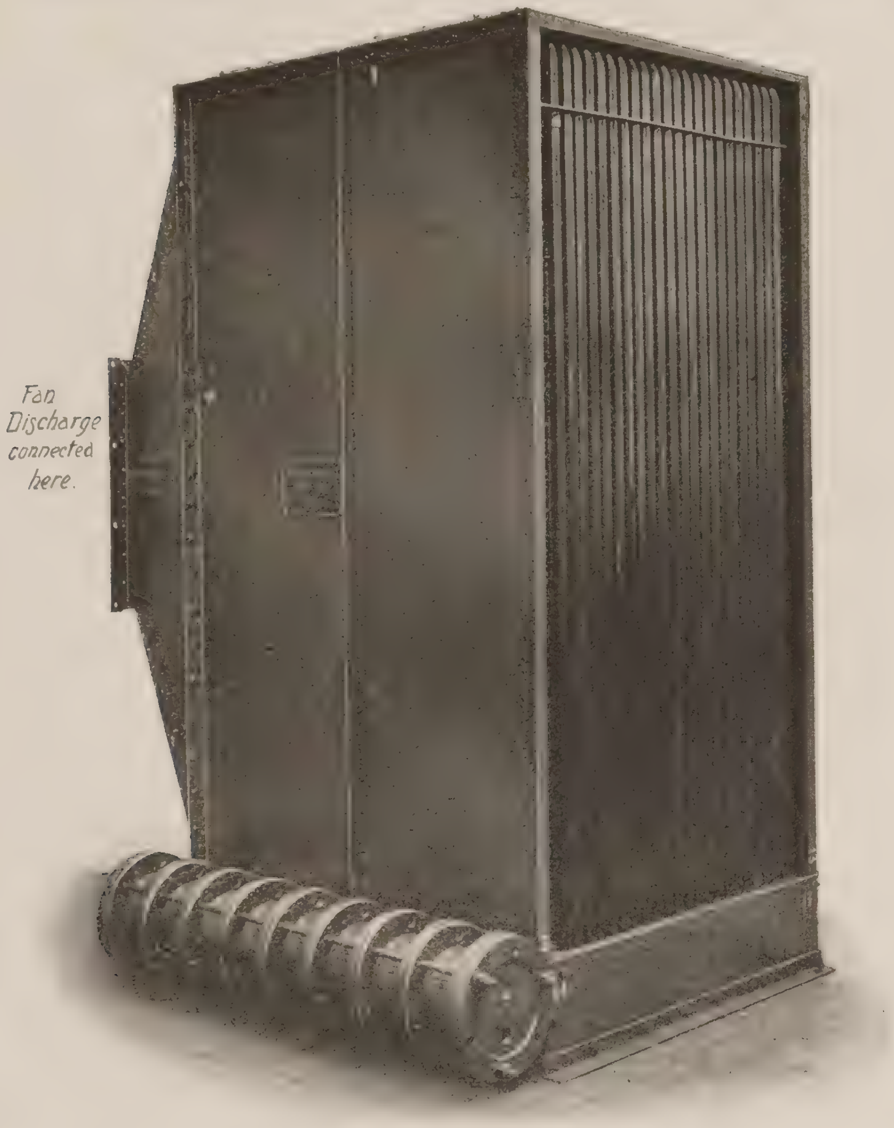


FIG. 17.—Weldless Steel Tube Heater for Steam. (By the courtesy of Messrs. Davidson & Co., Ltd.)

(*c*) Reduction in heating surface as compared with hot-water heaters, due to higher temperature (180° Fahr. hot water, 227° Fahr. steam).

It is obviously undesirable to generalise, for each installation must be carefully considered in conjunction with the

local circumstances, but, whether steam or water is employed for the heating medium, the heater is usually constructed on the same general method, the only difference being in size and positions of connections, air cocks, and area of surface.

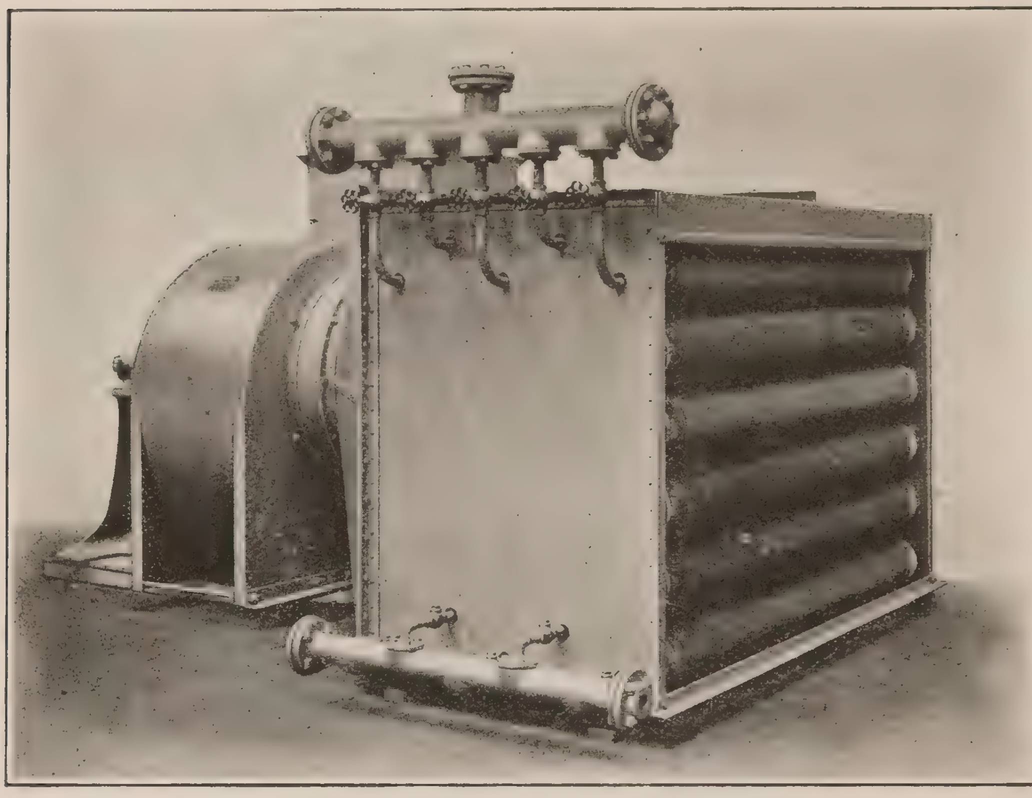


FIG. 18.—Gilled Cast-iron Pipe Heater. (By the courtesy of Messrs. Davidson & Co., Ltd.)

4. Types of Heaters.

The fundamental principles underlying the design of heaters are—

(*a*) To pack the maximum amount of heating surface into the minimum space.

(*b*) To set up minimum resistance to air flow.

Two chief types of heaters are in general use, viz. :—

(*a*) Pipe Heaters (steam only).

(*b*) Cast-Iron Heaters (steam and hot water).

The general construction of a pipe heater consists of a cast-iron box or header into which the two ends of the

wrought-iron pipes are secured. In some cases the header is divided internally into two compartments com-

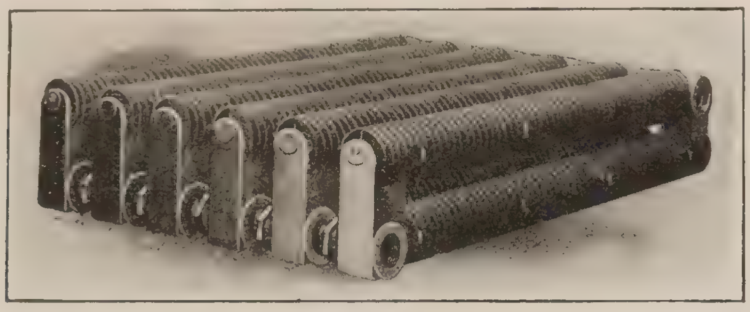


FIG. 19.—(*a*) "Excelsior" Cast-iron Sectional Heater, showing stack built up of six sections. (By the courtesy of the National Radiator Co., Ltd.)

municating directly with the inlet and drip (condense or exhaust) mains respectively. These are extensively used in the United States.

Other types have straight instead of hairpin-shaped

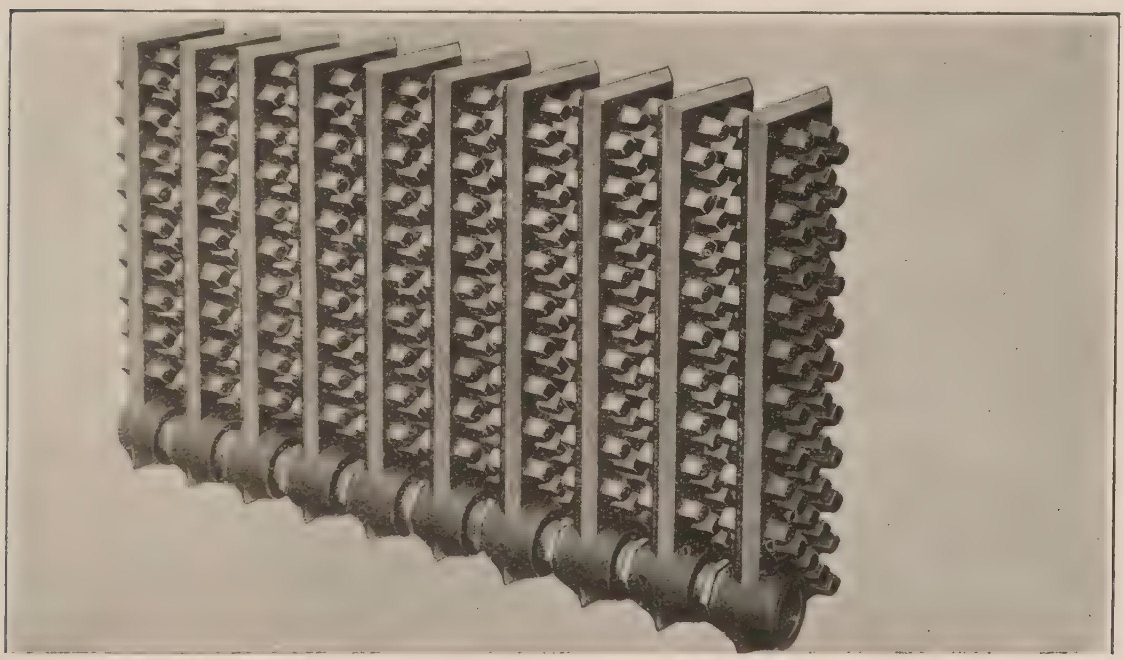


FIG. 19.—(*b*) "Vento" Cast-iron Sectional Heater, showing stack built up of ten sections. (By the courtesy of the National Radiator Co., Ltd.)

pipes, inlet headers of cast iron being provided at the upper and drip headers at the lower ends, into which the pipes are secured by expanding the ends or by means of lock nuts.

An improvement on this form is the cast-iron heater, constructed of stacks, built up of 4-inch cast-iron pipe

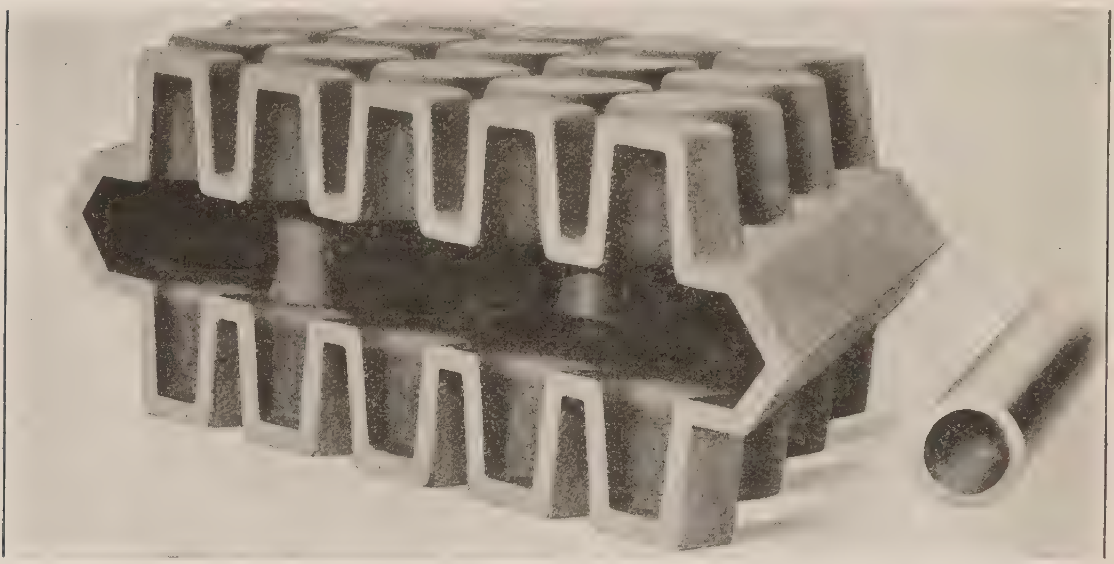


FIG. 19.—(c) Section of "Vento" Cast-iron Sectional Heater. (By the courtesy of the National Radiator Co., Ltd.)

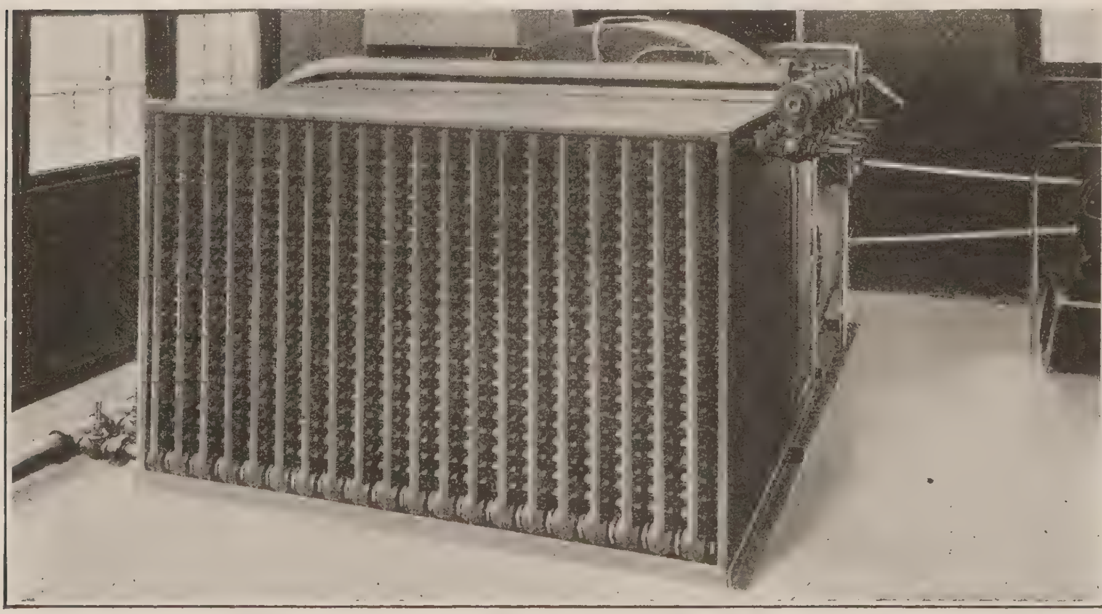


FIG. 19.—(d) Battery built up of "Vento" Sections. (By the courtesy of the National Radiator Co., Ltd.)

provided with radial gills or fins, which materially increase the effective area of the heating surface.

All cast-iron heaters are well adapted for use in connection with plenum ventilation work in difficult positions. The

complete heater is built up of small cast-iron units termed sections, which are assembled by means of right and left handed screwed nipples into stacks, and these are in turn coupled to cast-iron tee pipes, termed headers.

TABLE XXII.—**Surface and Free Area of "Excelsior" Cast-Iron Heaters** (National Radiator Co.).

Type.	Spacing of Centres (inches).	Free Air Space in per Cent. of Gross Area occupied.	Remarks.
Narrow .	$3\frac{3}{8}$	26	Heating surface is 12 sq. feet section. Dimensions $36\frac{3}{4} \times 8 \times 2\frac{1}{2}$ inches. Tapped for $1\frac{1}{2}$-inch pipe. Weight is 50 lbs.
Regular .	$3\frac{7}{8}$	**34**	
Wide .	$4\frac{3}{8}$	42	

Always use the "Regular" nipple whenever possible, in order to avoid delay in delivery. The maximum number of sections to be formed into one stack is limited by the quantity of heating medium (B.Th.U.) that can be efficiently passed through the $1\frac{1}{2}$-inch tapping.

For hot water figured on 20° drop, B.Th.U. at a velocity of 3 feet per second equals 162,000 B.Th.U.

For steam at 5 lbs. gauge at a velocity of 4,000 feet per second, B.Th.U. equals 139,000.

Friction drop of heating medium in its passage through the heater must be independently figured in conjunction with the head available (lbs. pressure per square inch for steam and feet of water column for a forced hot-water system.

A later and more efficient type is the "Vento" heater, also sold by the National Radiator Co., but chiefly used in America up to the present, of which the fullest possible performance information is published. This is a valuable feature when designing for closely priced and specified results), and a practice which is unfortunately uncommon with English

contractors and manufacturers, many of whom still continue to rate their apparatus in "length of 4-inch pipe," and then complain of foreign competition.

TABLE XXIII.—**Surface and Free Area of "Vento" Cast-Iron Heaters** (National Radiator Co., Ltd.).

Height of Section (inches).	Heating Surface per Section (sq. feet).		Spacing of Centres (inches).	Free Area per Cent. of Gross Area Occupied.	Remarks.
	Regular $9\frac{1}{8}$ inches wide.	Narrow $6\frac{3}{4}$ inches wide.			
40	10·75	7·50	$4\frac{5}{8}$	37	All tapped $2\frac{1}{2}$ inches Left and Right Handed. Weight is 8·2 lbs. per sq. foot of surface.
50	**13·50**	**9·50**	**5**	**44**	
60	16·00	11·00	$5\frac{3}{8}$	52	

It will be seen that numerous combinations can be made between the different heights, widths (regular and narrow), and spacings, either or all of which are interchangeable.

When steam and condense connections are on one side of the stack the number of sections per stack with a $2\frac{1}{2}$-inch tapping should be limited to 18; with steam and condense at opposite ends this may be increased to 30. In other words, not more than about 400,000 B.Th.U. per hour at 5 lbs. gauge and a pipe velocity of 4,000 feet per second should be taken from one stack under these conditions.

For hot water, as the sections are tapped $2\frac{1}{2}$ inches, figuring on a 20° Fahr. drop at a water velocity of 3 feet per second, the maximum safe heat possible per stack is 450,000 B.Th.U.

These cast-iron heaters, which occupy 15 per cent. less than a pipe coil of similar area, are well adapted for use in connection with plenum ventilation work, being extremely convenient to handle, absolutely interchangeable, admitting of reconstruction or extension of the system, with minimum

TABLE XXIV.—**Air Friction of "Vento" Cast-Iron Heaters (Inches W.G.).**

$4\frac{5}{8}$-inch, 5-inch, and $5\frac{3}{8}$-inch Centres. Regular (R.) and Narrow (N.) Sections. (National Radiator Co.).

Spacing Centres (inches).	Air Velocity through Free Area of Heater (feet per minute).													
	600.		800.		1,000.		1,200.		1,400.		1,600.		1,800.	
	R.	N.	R.	N.	R.	N.	R.	N.	R.	N.	R.	N.	R.	N.
$4\frac{5}{8}$	·022	·018	·040	·032	·063	·050	·090	·072	·122	·098	·160	·128	·202	·162
5	·021	·017	·037	·030	·059	·047	·084	·067	·115	·092	·150	·120	·190	·152
$5\frac{3}{8}$	·019	·015	·033	·027	·052	·042	·074	·061	·101	·083	·132	·108	·167	·137

Friction loss is in inches of water gauge *per stack*. For batteries containing more than one stack, multiply the appropriate loss for one stack (obtained from Table XXIV.) by the number of stacks deep in the direction of air flow, to obtain the total loss of static head.

Some suggested velocities for air flow through the free area of heaters are given in Table XXVI.

scrapping of old parts, are free from rust troubles, and are easy to replace in the remote event of failure of a section.

5. Velocity Through and Loss of Pressure in Heaters.

Air in passing through heaters of any type sets up frictional resistance, the value depending on the velocity of the air, the depth of the heater in the direction of air flow, and the finish (smoothness or otherwise) of the surface.

TABLE XXV.—**Air Friction of Pipe Heaters (Inches W.G.).** One-inch coils on $2\frac{3}{4}$-inch Centres (Thompson).

Air Velocity through Free Area (feet per minute).	Number of Stacks deep in Direction of Air Flow.			
	4 Rows.	8 Rows.	12 Rows.	16 Rows.
600	0·04	0·06	0·09	0·15
800	0·06	0·10	0·15	0·19
1,000	0·09	0·15	0·21	0·30
1,200	0·12	0·21	0·31	0·43
1,400	0·17	0·30	0·45	0·60

TABLE XXVI.—**Suggested Air Velocity Values for Various Heaters** (through Free Area of Heater : W. H. Carrier).

No. of Stacks deep in Direction of Air Flow.	Total Fan Pressures (Inches of Water).						
	Public Buildings (Quiet running).				Industrial Plant (probably more or less noisy).		
	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$
4	990	1,140	1,280	1,400	1,510	1,610	1,800
5	885	1,020	1,140	1,250	1,350	1,440	1,610
6	810	930	1,040	1,140	1,230	1,320	1,470
7	745	860	960	1,055	1,140	1,220	1,360
8	700	810	910	995	1,070	1,150	1,280

The table is based on the assumption that the pressure loss through the heater shall not exceed 50 per cent. of total static pressure set up by fan. Variations above and below these values are of course common in every-day practice. In fact, the whole design is usually one of compromise between the model conditions desired by the engineer and impossible cubic space provided by the designer or owner of the building.

6. By-Passes.

In order to obtain rapid variations of temperature and materially to reduce the frictional losses caused by the heater, when it is not required to warm the air, dampers should be provided under both the tempering and re-heating coils, so that air can pass straight to the fan and on to the washer and ducts without passing through the larger part of the heater.

Frequently the dampers are made equal in area to not less than 10 or more than 15 per cent. of the *gross* area of the heater.

When dampers are larger than 20 inches by 36 inches they should be made of the louvred pattern to reduce the action of the air on them as much as posssible, and especially is this necessary if they are to be automatically controlled.

7. Surface Efficiency of Heaters.

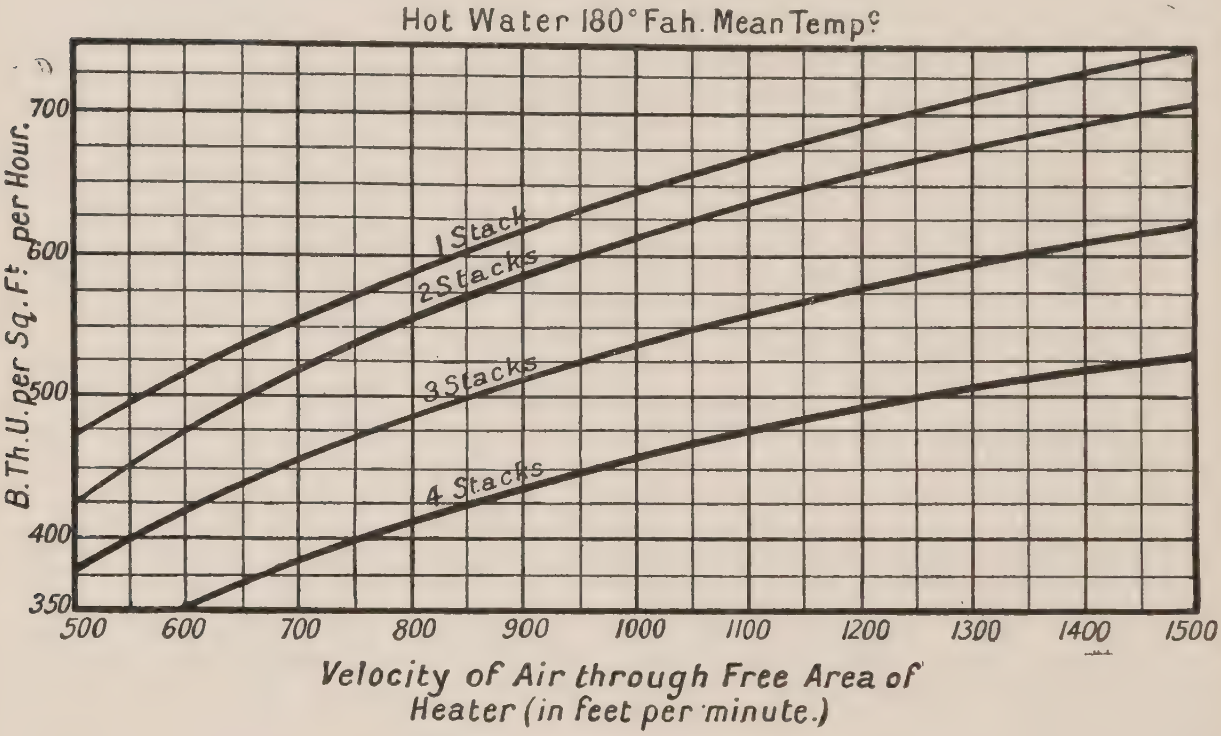
The efficiency of any plenum heater depends on two factors—

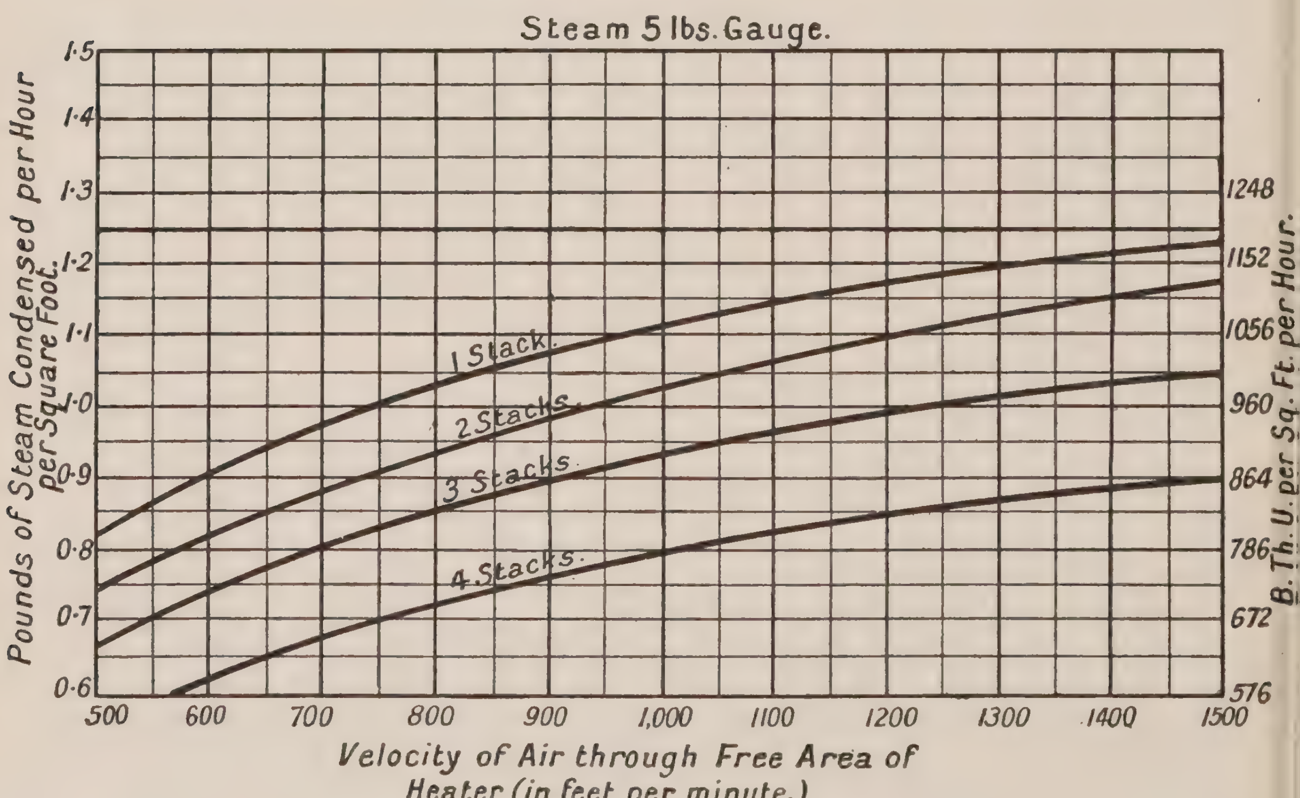
(*a*) The velocity of air through the heater.

(*b*) The difference in temperature of the air and the heating surface (which in the case of steam heating depends to a small extent on the pressure).

The velocity of air through a heater is limited by the head of air pressure (static head) available to compensate for the friction loss in this part of the air circuit, and in this connection it will be remembered that friction loss is proportioned to the square of (not directly as) the velocity.

TABLE XXVII.—**Heating Surface Efficiency Curves.** (By the courtesy of the National Radiator Co., Ltd.)





Usually in public building ventilation the velocity through the free areas of the heater (which with Regular "Excelsior" sections on $3\frac{7}{8}$-inch centres is 34 per cent. of gross cross-sectional area, and with "Vento" sections 50 inches high on 5-inch centres is 44 per cent. ditto) is of the order of 1,200 feet per minute; but in certain cases the writer has been forced to work up to 1,600 feet per minute, with correspondingly high friction losses and running costs for electric current, when space has been extremely limited.

The depth of the heater in the direction of air flow depends upon the total heating duty required. Obviously, after the air has passed over the first few stacks it approaches more nearly the temperature of the heating surface; hence the transmission is reduced and additional stacks have not a proportionate heating value, so that it is exceptional to design a battery of more than five or six cast-iron stacks deep in the direction of air flow.

The efficiency curves clearly show—

(*a*) The reduced efficiency of the heating surface of additional stacks.

(*b*) Increased efficiency of heating surface at higher air velocities.

Under similar conditions, but with water at a mean temperature of 180° Fahr., the total of B.Th.U. transmitted per hour is about 75 per cent. of that with steam at 5 lbs. gauge (227° Fahr.).

To obtain the same increase in temperature with incoming air at same temperature and other conditions equal, the velocity of air will have to be reduced to 40 per cent. of that figured for steam 5 lbs. gauge (227° Fahr.).

8. Quantity of Heating Medium Required.

The quantity of heating medium, either hot water or steam, required for conditioning the air can be rapidly obtained if heat values are all reduced to the B.Th.U. basis instead of utilising the hopelessly antiquated and misleading "square feet of radiating surface," or, worse still, the "equivalent length of 4-inch pipe" rating.

TABLE XXVIII.—**Water Required for Heater with Mechanical Circulation.**

(Flow, 190° Fahr. Return, 170° Fahr. Mean temperature of water, 180° Fahr.)

	British Thermal Units per Hour.						
	50,000	100,000	250,000	500,000	1,000,000	2,500 000	5,000,000
Boiler horse-power	1·6	3·3	8·3	16	33	83	166
Gallons of water per hour . . .	250	500	1,250	2,500	5,000	12,500	225,000
Gallons of water per minute . .	4·1	8·2	20	41	82	200	410

The British Thermal Unit is defined as "That quantity of heat which is required to raise one pound of water through one degree Fahrenheit" (strictly speaking, from 32° to 33° Fahr.). It has been also shown (Chapter I., par. 8) that it will also raise 53 cubic feet of air through the same limits of temperature.*

Considering a hot-water system, the maximum working temperature (to avoid joint trouble due to excessive heating or baking of the jointing compound and also generating steam in the boiler) is 190° Fahr., and in order to obtain reasonably consistent results from all radiators the velocity of flow of water, and hence pipe sizes, are first figured on a 20° drop or difference between flow and return, giving a mean temperature in the system of 180° Fahr. One gallon of water weighs (for practical purposes) 10 lbs., and therefore each gallon contains for heating purposes:—

$$10 \text{ lbs.} \times 20° \text{ Fahr.} = 200 \text{ B.Th.U.}$$

EXAMPLE 15.—Therefore, if a heater be divided into ten stacks of two tiers, *i.e.*, five stacks on the ground and five ditto immediately above, water required would be:—

(*a*) *For heater*:—

$$\frac{1{,}000{,}000}{200} = \begin{cases} 5{,}000 \text{ gallons per hour, or} \\ 84 \quad ,, \quad ,, \text{ minute.} \end{cases}$$

(*b*) *Per stack*:—

$$\frac{1{,}000{,}000}{10 \times 200} = \begin{cases} 500 \text{ gallons per hour, or} \\ 8{\cdot}4 \quad ,, \quad ,, \text{ minute.} \end{cases}$$

In figuring steam systems the heat utilised is that which was absorbed when the water was evaporated into steam, termed the "Latent Heat of Vaporisation (Steam)," or, briefly, "Latent Heat," and which is available when the steam is again condensed to water, the condense being returned to the boiler as hot as possible, to be again converted into steam.

In low-pressure steam heating work the gauge pressure adopted is generally 5 lbs. per square inch, the latent heat

* One boiler horse-power is equivalent to 30,000 B.Th.U.

TABLE XXIX.—**Steam Required for Heater.**

	British Thermal Units per Hour.						
	50,000	100,000	250,000	500,000	1,000,000	2,500,000	5,000,000
Boiler horse-power	1·6	3·3	8·3	16	33	83	166
Lbs. of steam per hour . . .	52	104	260	520	1,040	2,600	5,200
Gallons of condensed water per hour .	5·2	10	26	52	104	260	520

in B.Th.U. per lb. of steam corresponding to this pressure being approximately 960, and the temperature of the steam 228° Fahr. (For details of "Atmospheric," "Vacuum" and other systems, further information on latent heat of steam, etc., the reader is referred to text-books dealing exclusively with the subject of heating).

Therefore, considering the hot-water problem dealt with on page 89—

EXAMPLE 16.—Steam required :—

(*a*) For *heater*, $\frac{1,000,000}{960} = 1,040$ lbs. steam per hour.

(*b*) Per *stack*, $\frac{1,000,000}{10 \times 960} = 104$,, ,, ,, ,,

9. Arrangement and Casing of Heaters.

The control of the final temperature of the air and the regulation of the humidity on steam heaters is effected either by reducing or increasing the amount of heating surface in use or by mixing cold air with the heated air in varying proportions to obtain the desired results. In hot-water heaters there are the additional alternatives of varying the temperature or the quantity of water.

Of these the variation of surface and the damper methods are most popular. The cast-iron heaters are made up of individual units called sections, assembled by means of right and left hand threaded nipples into larger units called stacks, and these are fed from a header or main pipe.

In pipe heaters wrought-iron pipes are fixed into the cast-iron boxes called sub-headers forming stacks, and cast-iron heater sections are also formed in stacks, both of which in turn will be connected to the feed pipe, called the main header. Successive stacks should be staggered, *i.e.*, the sections of one stack placed opposite the spaces of the other, so that the air takes a zig-zag course through the heater and thus becomes thoroughly heated.

Each of the stacks should be connected through separate valves, on both flow and return, on to the main flow and return

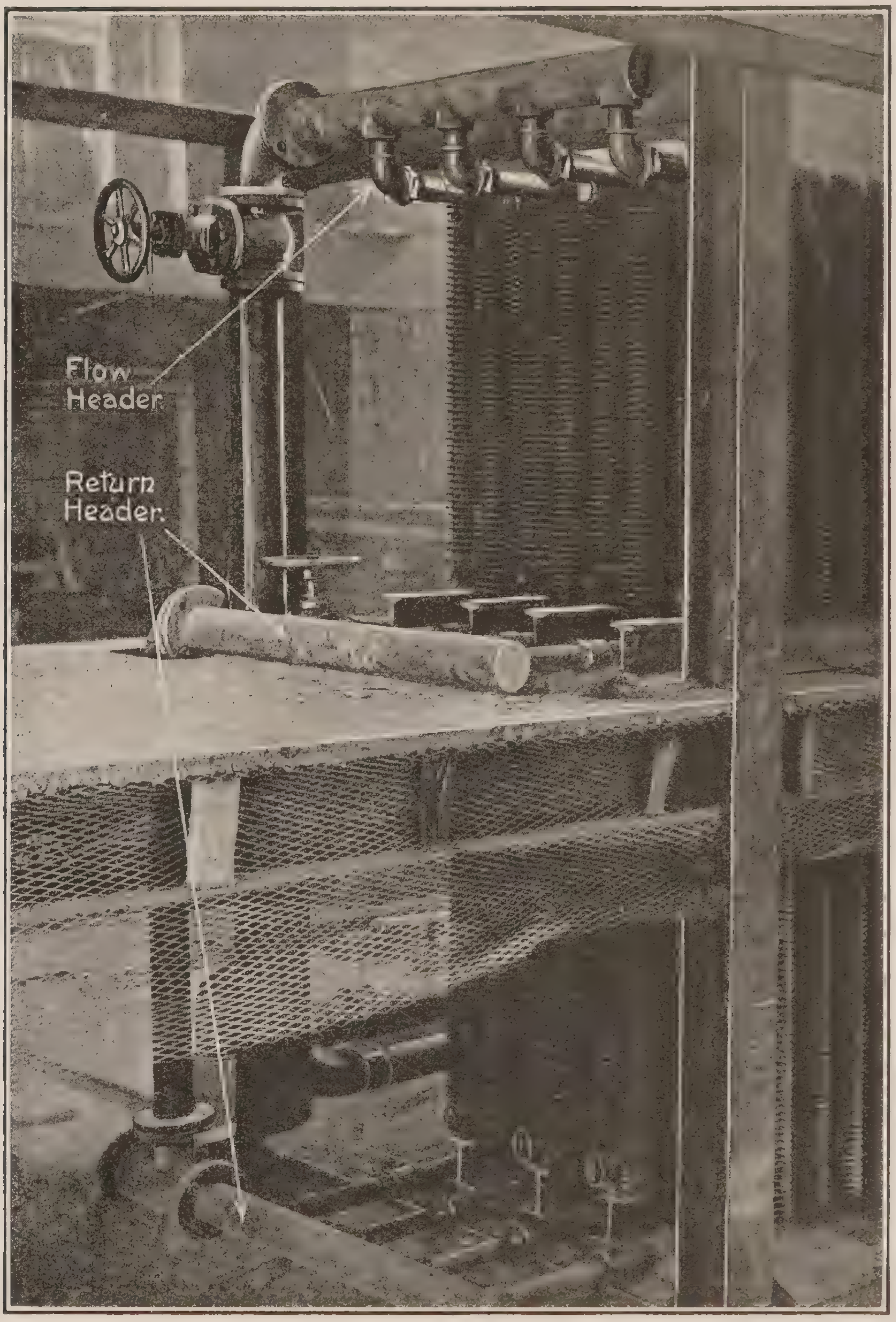


FIG. 20.—Plenum Battery for conditioning $2\frac{1}{2}$ million cubic feet of air per hour. (Constructed with "Excelsior" cast-iron sections of the National Radiator Co., Ltd.)

headers, so that on mild days only one or two of them are used and on cold days all stacks are opened up.

Heaters are usually enclosed in a sheet-iron case of 20 B.W.G. up to $\frac{3}{16}$-inch plate, stiffened by $1\frac{1}{2}$-inch angle

iron frame work. Frequently brick or concrete walls, floor, and ceiling form three sides of a heater, sheet iron which is easily removed for inspection or repairs being added to enclose the fourth.

Heaters are frequently supported either upon light T irons $\frac{3}{8}$-inch pipe, $\frac{1}{2}$-inch bar resting upon a brick foundation, or 8-inch channel iron, 4-inch by 3-inch R.S.J., etc.

Connections between the fan and heater casing should be carefully designed to avoid undue loss of air pressure in converting from static to velocity head.

If the angle between the connection and the side of the fan is too acute, a considerable portion of the heater may be "dead."

TABLE XXX.—**Length of Heater Connections to Fan** (Carrier).

Capacity of Fan (Cubic Feet of Air per Minute).	Minimum Distance from Fan to Heater.*
6,000	1 ft. 6 in. to 2 ft.
8,000—16,000	2 ft. to 2 ft. 6 in.
16,000—27,000	3 ft.
27,000—47,000	3 ft. 6 in.
47,000—65,000	4 ft. to 4 ft. 6 in.

* Desirable to adhere to 12° to 30° rule.

10. Automatic Control.

It is useless sinking considerable sums of money in equipping a building with a first-class ventilating plant, capable of maintaining ideal atmospheric conditions and yet failing to provide for the efficient operation of it, a practice unfortunately all too common in this country, and one which has been largely to blame for the many "failures" recorded.

Attendants with brains are difficult to find, cost money, and even then constant attention is necessary to forecast, to follow and allow for all the vagaries of our climate. This

being so, automatic regulation should be adopted wherever funds allow, and the brawn (not brain) work of firing the boilers and cleaning of plant only, left to the attendant.

Automatic control apparatus is divided into two parts, *i.e.*—

(*a*) The instruments which sample the conditioned air.

(*b*) The valves which control the supply of heating medium to the various parts of the apparatus, or the position of dampers, and are in themselves actuated either directly or indirectly by the sampling instruments as required.

Sampling instruments are termed "thermostats" when affected by the temperature of the dry bulb, and "humidostats" when effected by the temperature of the wet bulb or by the relative humidity of the air.

The tendency now is to combine temperature and humidity control, and various types of apparatus, some more or less of an experimental nature, are offered by several makers.

There are three variables in connection with automatic temperature and humidity control, *i.e.*—

(1) Temperature of air leaving tempering coil (2) Temperature of spray water	affecting weight of moisture absorbed.
(3) Temperature of air leaving reheating coil	

Generally speaking it is the latter which affects the comfort of the apartment most directly, and is therefore looked upon as the basis of operations. A dry bulb thermostat is therefore installed somewhere in the ducts where a fair average sample of the air leaving the conditioning apparatus can be obtained, which controls the heating medium supplied to the re-heating coils, maintaining the dry bulb temperature at 65° Fahr.

In par. 2 of this chapter it was stated that the conditioned air should be at 65° Fahr. dry bulb temperature and 43 per

cent. relative humidity, which corresponds to a wet bulb depression of 12° Fahr. It will thus be obvious that if some form of humidostat or wet bulb thermometer is also placed in the conditioned air duct and connected to a supply of heating medium which will directly affect the temperature of the spray water, *i.e.*, the tempering coils or heating surface placed directly in the settling tank of the air washer, the humidity will be automatically controlled, for any increase or decrease in the ducts will result in more or less heat being supplied to the spray water, either directly or indirectly, and hence to a correction of the humidity. When heating coils are placed in the settling tank a further thermostat should be fixed to take the temperature of the external air and admit heating medium to the tempering coils in hard weather.

Control apparatus may be classified into two types, viz.—

(*a*) Sampling instruments which supply the energy from within themselves to operate the valves on the heat supply or the by-pass damper.

(*b*) Sampling instruments which merely control the motive power operating the valves on the heat supply or by-pass dampers.

The present writer prefers to control the supply of the heating medium rather than to alter the positions of dampers, as the latter are generally badly placed as regards air flow, causing whirls, eddies, and other losses, and are expensive to design for accurate balance, which is essential if they are to be operated by lower power.

Thermostats and humidostats are dealt with more fully in Chapter IX.

Unfortunately the whole subject, as is the case with the majority of details connected with ventilating plant, has been sadly neglected by English engineers, and we are therefore, with one or two exceptions, dependent on the United States imported apparatus for carrying out this important work.

11. Pipe Friction and Fittings, Boilers, and Flues.

In order to keep the size of the present volume within practical limits it is considered undesirable to discuss here in detail accurate pipe sizes, heat losses from lagged and unlagged pipes, pipe fittings and design of pipe-work methods of accelerating flow in hot-water systems and

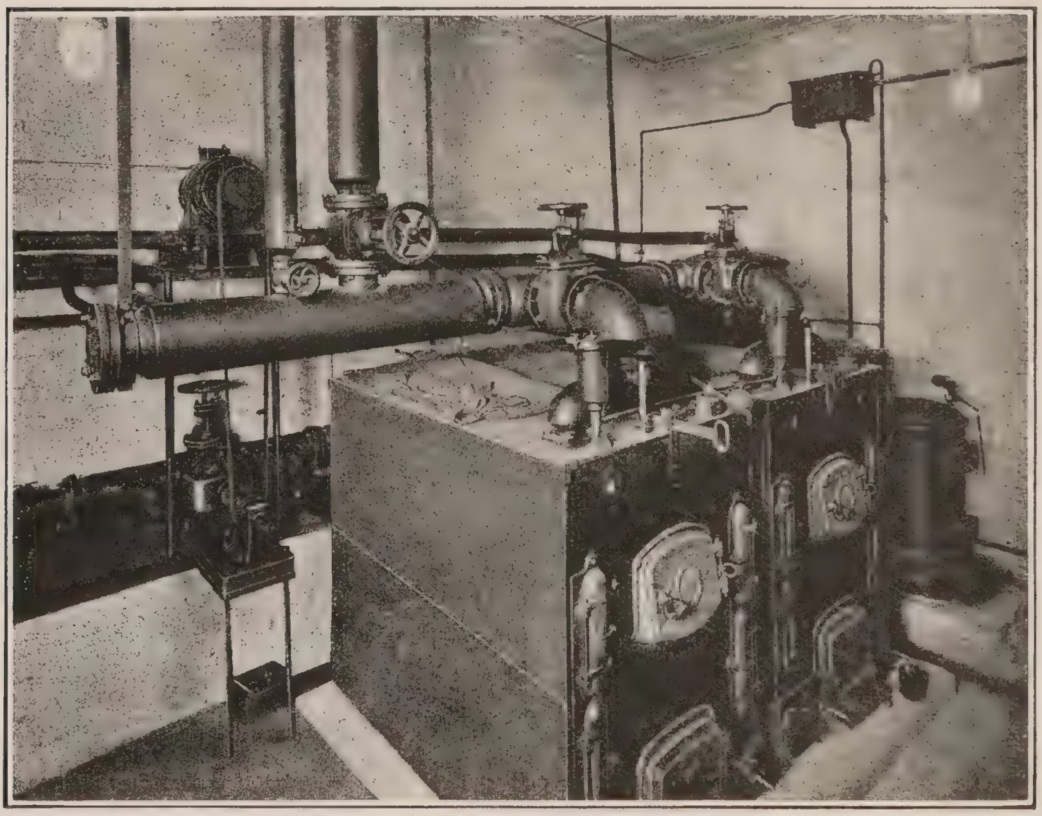


FIG. 21.—Battery of Cast-iron Sectional Low Pressure Hot Water Boilers, for a duty of 2,000,000 B.Th.U. per hour on the five-hour stoking rating (National Radiator Co.).

returning condense to boilers in steam systems, steam traps, feed tanks and pumps, air valves, boilers, automatic regulators, safety valves, thermometer, steam, and altitude gauges, etc., etc. Readers desiring further information on these points, which are of course closely connected with ventilating plant, are therefore courteo sly referred to books dealing exclusively with the subject.

The efficiency of any air condit oning installation depends directly upon the efficiency of the heater, and th s in turn

TABLE XXXI.—**Areas and Heights of Chimney in Feet** (National Radiator Co.).

Area of Flue (sq. inches).	Height of Chimney (feet).				
	25	40	55	70	85
36	80,000	100,000	115,000	130,000	145,000
49	105,000	130,000	150,000	170,000	190,000
64	135,000	170,000	200,000	230,000	260,000
81	170,000	220,000	255,000	290,000	325,000
100	210,000	275,000	315,000	355,000	395,000
121	270,000	350,000	400,000	450,000	500,000
144	305,000	390,000	450,000	510,000	570,000
162	350,000	440,000	510,000	580,000	650,000
196	420,000	530,000	620,000	700,000	780,000
225	485,000	610,000	710,000	800,000	890,000
252	540,000	680,000	800,000	900,000	1,000,000
289	620,000	780,000	920,000	1,030,000	1,140,000
324	700,000	880,000	1,030,000	1,160,000	1,290,000
361	770,000	980,000	1,140,000	1,280,000	1,420,000
400	860,000	1,090,000	1,260,000	1,420,000	1,580,000
576	1,235,000	1,500,000	1,750,000	2,000,000	2,250,000

For well-constructed flues the above dimensions may be reduced by 10 to 20 per cent.

The figures given under the varying heights of chimneys represent the boiler capacities in B.Th.U. per hour. A square or round flue is to be preferred, but rectangular shapes may be used if the area is equal and the difference between width and length is not extreme. (See Table XIV., Chapter IV., " Equalisation of Area for Round and Rectangular Pipe.")

on the boiler and on the boiler flue, all of which must be of ample capacity.

Of the total heat available in coke or anthracite approximately 8,000 B.Th.U. per lb. weight consumed in the boiler is available in the heating medium.

Air supply is required for complete combustion at the rate of 20 to 25 lbs. per lb. of fuel burned, say, 270 cubic feet measured at 70° Fahr.

Hence the total quantity of air required is :—

$$\frac{\text{Total boiler-power per hour in B.Th.U.} \times 270}{8{,}000 \times 60} = \text{cubic feet}$$

of air per minute.

As regards the gases to flue, these will be at a much higher temperature, probably 450° Fahr., and hence volume will be 1·7 as great, viz., 460 cubic feet per lb. weight of fuel burned.

Therefore volume of flue gases measured at 450 (omitting weight of fuel, which is only about 4 per cent. of total) is :—

$$\frac{\text{Total boiler-power per hour in B.Th.U.} \times 460}{8{,}000 \times 60} = \text{cubic feet}$$

of gases to flue per minute.

Draught intensity of a chimney is proportiona to its height. On the other hand, draught losses vary as the square of the velocity ; hence the necessity of figuring on low speeds, both for the air supply ducts to boiler-house and for the flue gases, if natural draught is adopted. To avoid extended calculation for the latter, Table XXXI. is appended.

CHAPTER VIII

OZONE

1. Properties.

Many experiments have been made and much thought devoted to the problem of ascertaining wherein lies the fundamental difference between the atmosphere of a bracing and invigorating seaside town and a so-called relaxing and depressing inland town under similar conditions of barometer, thermometer, and hygrometer.

Oxygen, of course, has a stimulating effect on the nervous system, being frequently used by medical men in cases of severe prostration, but this gas is usually found in almost equal proportions in all localities.

About 1785 chemists observed that there were two forms of oxygen, one of which had slightly higher oxidising properties than the other, and to this latter, which is a peculiarly strong-smelling substance, was given the name of *Ozone*.

The conversion of oxygen into ozone is expressed chemically by the equation

$$O_2 + O = O_3,$$

or Molecular Oxygen + Nascent Oxygen = Ozone.

This is an unstable compound, the third atom readily combining with other bodies to form oxides, thus rendering it a powerful sterilising agent.

Analysis has shown repeatedly that ozone, which is probably formed in nature by the action of electric discharges during thunderstorms, the silent discharges from clouds, the evaporation of saline water, as in sea foam, or the action of sunlight on clouds, is found in greater quantities in the open country and at the seaside than in manufacturing towns, where it is either absent or only exists in minute and practically untraceable quantities.

This phenomenon is of course to be anticipated in view of the comparatively large quantities of carbon, sulphur, phosphorus, and other undesirable compounds liberated from factory chimneys.

When ozone is manufactured artificially in a pure state and introduced into the fresh air ducts in the proportion of 0·2 to 1·0 parts in one million parts of the entering air (London tube railways 0·3 to 0·4 part) it imparts an agreeable and refreshing odour, and is said to produce an invigorating effect similar to that experienced during a visit to a bracing resort, although the benefits or otherwise to be derived from the use of ozone, necessarily in such minute quantities, are still a matter of discussion. Ozone has also proved of value in assisting to neutralise smells in kitchens, urinals, underground *cafés*, smoking rooms, and manufacturing processes, but in the latter case the concentration is considerably stronger. It is generally held that ozone should prove of considerable value in ventilating systems designed to re-use re-circulated air, but in this, as in the question of the essentials of ventilation, engineers are still awaiting a definite lead from the chemists and physicists.

2. Methods of Production.

Whilst there are several chemical methods of producing ozone, they are quite unsuitable as at present developed for its manufacture on a commercial scale, and, since high tension electrical discharges are always accompanied by the production of ozone, it is this method which has been recently developed.

Great care has to be exercised in the design of suitable apparatus to avoid the production of oxides of nitrogen, which irritate the membrane of the throat, and also to avoid high temperatures in the apparatus, for the compound disintegrates at a temperature of 100° Centigrade into oxygen again.

These differences are overcome by utilising the principle of a silent alternating current electrical discharge across a dielectric which fixes the width of the gap. The efficiency

is also reasonably high, being of the order of 500 grammes per kilowatt hour; hence the cost of operation for ordinary use is negligible.

3. Apparatus Employed.

The ozoniser as employed by Ozonair, Ltd., consists essentially of two metal gauze electrodes or conductors,

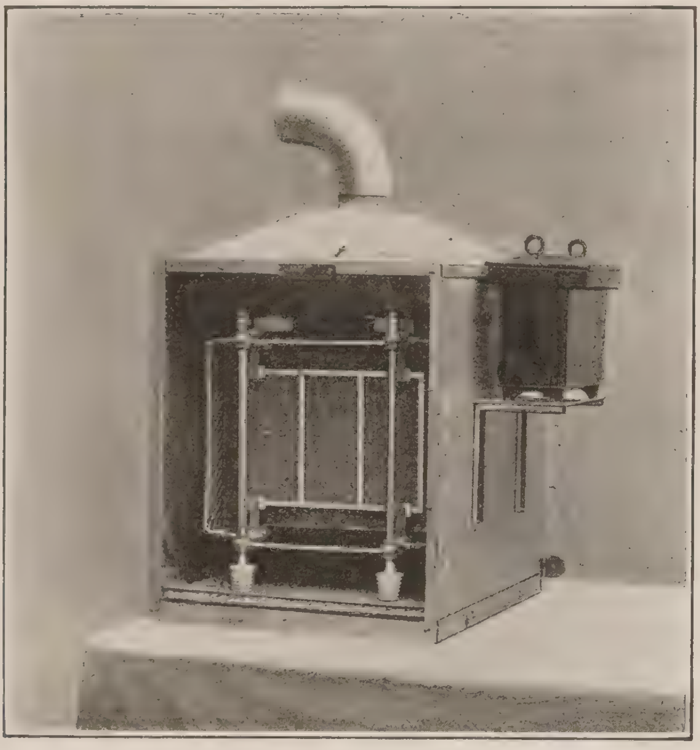


FIG. 22.—Ozone Apparatus. (By the courtesy of Messrs. Ozonair, Ltd.)

separated by a non-conductor or dielectric. The two electrodes, acting either as a complete unit or built up with others to form larger units for greater output, are held in position by clamps of non-conducting material, protected by strongly meshed wire, and enclosed in a galvanised iron case.

At the bottom of the case is a removable frame which contains a layer of absorbent wool used to filter the air passing through the ozoniser. The suction pipe frequently communicates with the fresh air and the delivery pipe to the

intake to the fan, thus utilising the suction pressure of the fan to draw the required amount of air through the apparatus.

Electricity on the alternating current single-phase system at a pressure of 6,000 to 9,000 volts, is supplied from the small static transformer fixed at the side of the case to the electrodes, causing the silent discharge and the production of ozone.

The static transformer can be wound for any alternating current supply voltage and periodicity. When a continuous current only is available a small rotary converter for producing alternating current has to be added, but this is not a serious affair, as the size is only about 2 feet by 2 feet by 2 feet.

Regulation of the intensity of ozone is obtained in five steps by a small switch connected to the low-pressure winding of the transformer or in the direct-current side of the converter.

4. Bibliography.

Interesting contributions on the subject are: "Air Ozonisation" (Franklin), "Odors and their Composition" (Allen), "A Reply to Recent Comments on Air Ozonisation" (Franklin), "An Experiment with Ozone as an Adjunct to Artificial Ventilation" (*Heating and Ventilating Magazine*, October, 1914, and March, 1915).

CHAPTER IX

INSTRUMENTS

1. Short Range Thermometers.

The limits of temperature of any indoor apartment used for living purposes in this country will seldom exceed a minimum of 40° Fahr. or a maximum of 90° Fahr., and it is therefore unnecessary and inconvenient to use thermometers having scales graduated from — 30° through zero up to 250° Fahr., which, strange to say, are quite common. Further, thermometers which are frequently consulted should be easily read and reasonably accurate. Brass scales tarnish and rapidly become useless. The present writer therefore specifies a black spirit thermometer, having white celluloid scale, engraved with black figures, the limits being 40° Fahr. to 90° Fahr., giving an excellent open scale.

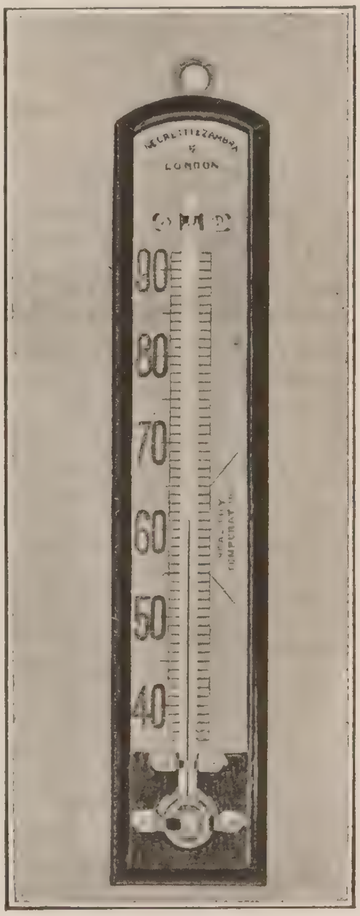
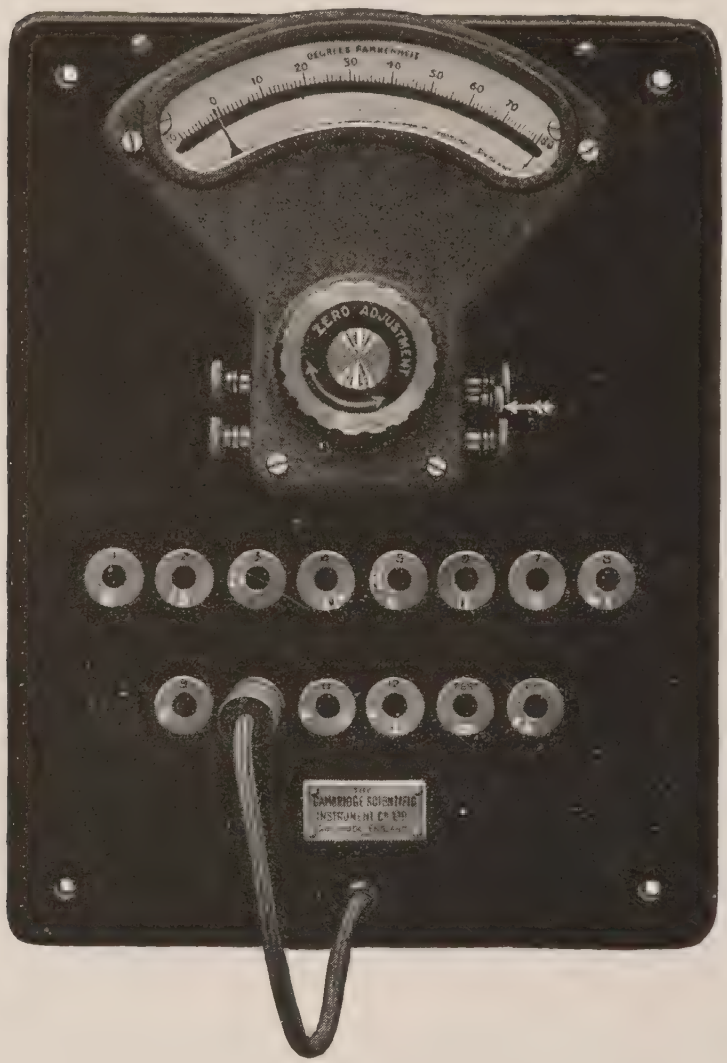


FIG. 23. — Short Scale Spirit Thermometer. (By the courtesy of Messrs. Negretti and Zambra.)

While not quite so accurate as the mercurial type, the advantages more than balance any very slight error in this respect.

In very large ventilating installations, in which it would require almost the whole time of one man to inspect the thermometers fixed in the various apartments, an extremely useful instrument s the Distance Thermometer, whereby the temperature of



(*a*) Instrument and Plugboard for connecting to different circuits.

(*b*) Thermometer Bulb containing platinum wire resistance.

FIG. 24.—Distance Thermometer and Switchboard. (By the courtesy of the Cambridge Scientific Instrument Co., Ltd.)

any apartment, regardless of distance, can be read on a single instrument placed at a convenient point, such as the resident engineer's office, the boiler-room, etc.

The principle upon which these instruments are based is very simple, and depends upon the fact that the electrical

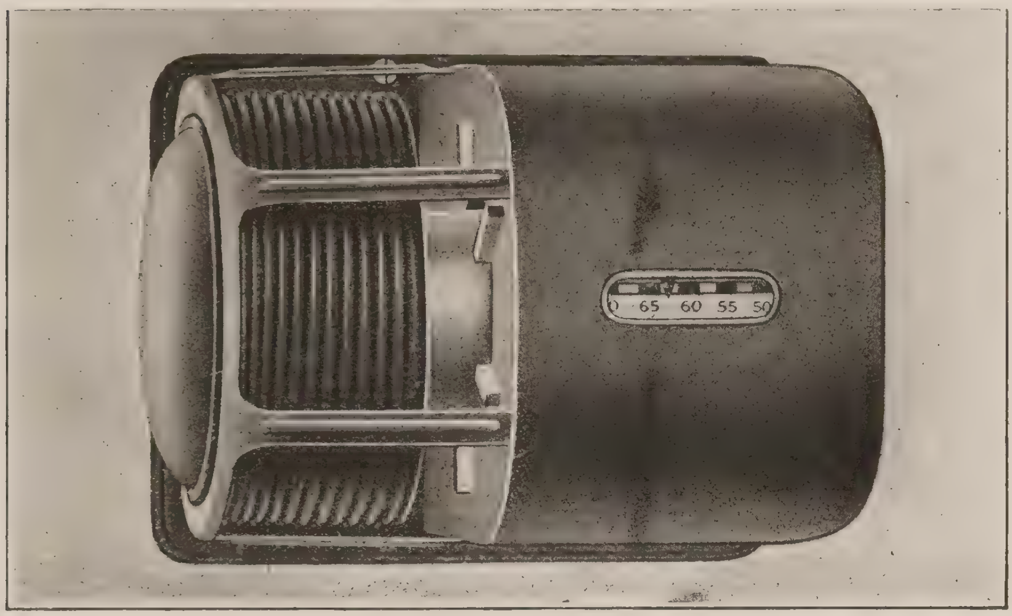


FIG. 25.—(*a*) Regitherm complete. (By the courtesy of the National Radiator Co., Ltd.)

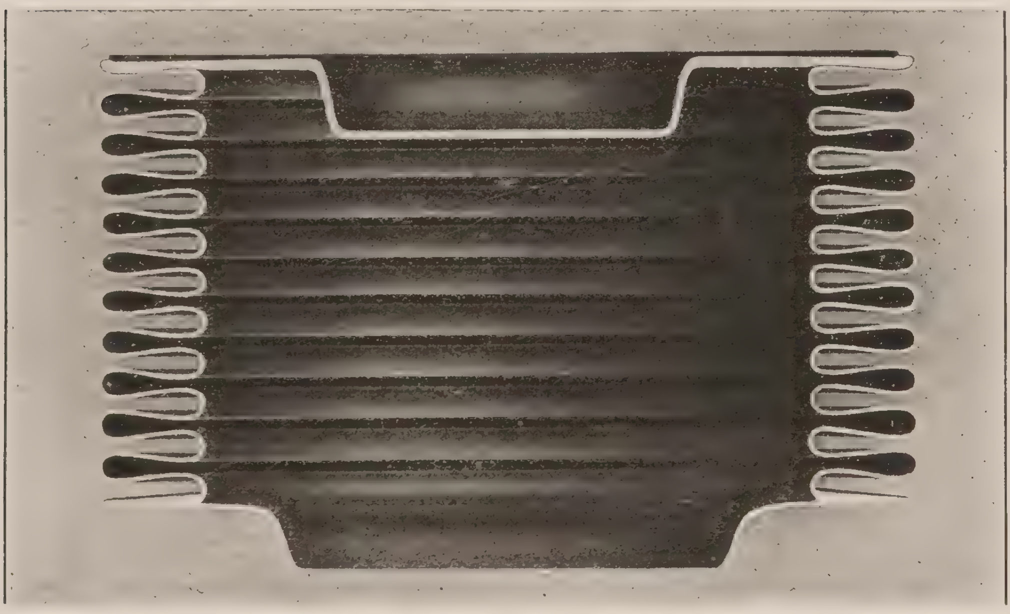


FIG. 25.—(*b*) Regitherm. Details of Metal Bellows. (By the courtesy of the National Radiator Co., Ltd.)

resistance of a platinum wire varies with its temperature in accordance with a well-known law.

By means of the "Wheatstone Bridge" principle of balancing electrical resistances the resistance of the platinum

wire is measured merely by inserting a plug in a hole on the switchboard, corresponding to the apartment the temperature of which it is required to ascertain, and reading the equivalent of the resistance from the instrument directly in degrees Fahr. Connecting wires consist of ordinary good quality $\frac{1}{16}$ or $\frac{3}{25}$ electric bell or telephone wire, the current required being supplied from a small two-volt accumulator, an adjusting screw on the instrument being provided to compensate for variations in voltage of the accumulator.

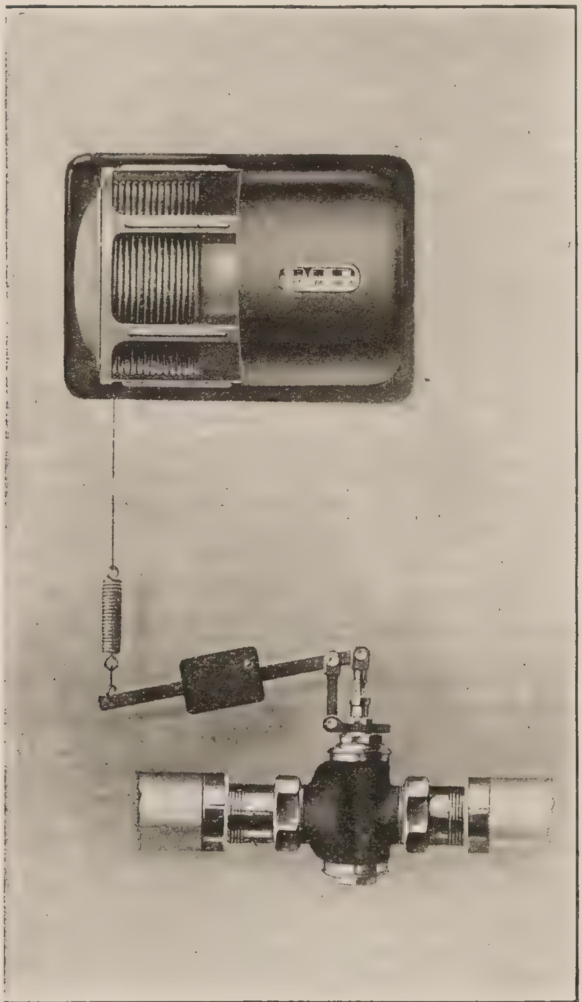


FIG. 25.—(*c*) Regitherm connected up to balanced valve. (By the courtesy of the National Radiator Co., Ltd.)

2. Thermostats.

There are various mechanical forms of thermometer on the market which depend for their action upon the well-known fact that many metals and liquids expand with increase of temperature.

This principle is utilised in the various forms of automatic temperature control apparatus now available, in which the expansion of a metal is utilised either to close or open a valve, governing the flow of compressed air or water, or to close an electric circuit, or to operate a series of levers connected to dampers, etc.

Provision is usually made for controlling the amount of movement, frequently by an adjustable spring, so that the

instrument can be set to operate at any temperature desired within its range.

The Regitherm is one of the direct operating type, the operating mechanism of which contains a very volatile fluid, which under the action of heat vaporises and exerts considerable pressure on a diaphragm, which is free to rise and hence operate levers or wires connected to dampers, valves, etc.

The "Johnson" thermostat is one of the type using compressed air, which operates valves, dampers, etc., by controlling the flow of air at a pressure of 15 lbs. per sq. inch. By the action of heat on the compound metal strip fixed at the bottom of the instrument the orifice is opened or closed, the flow of air to the diaphragm-operated apparatus being affected accordingly.

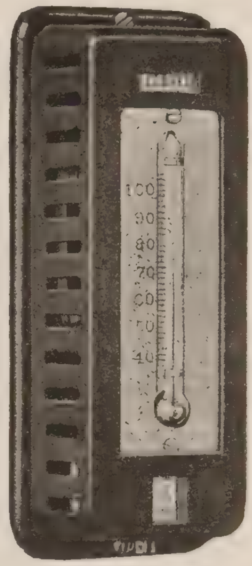
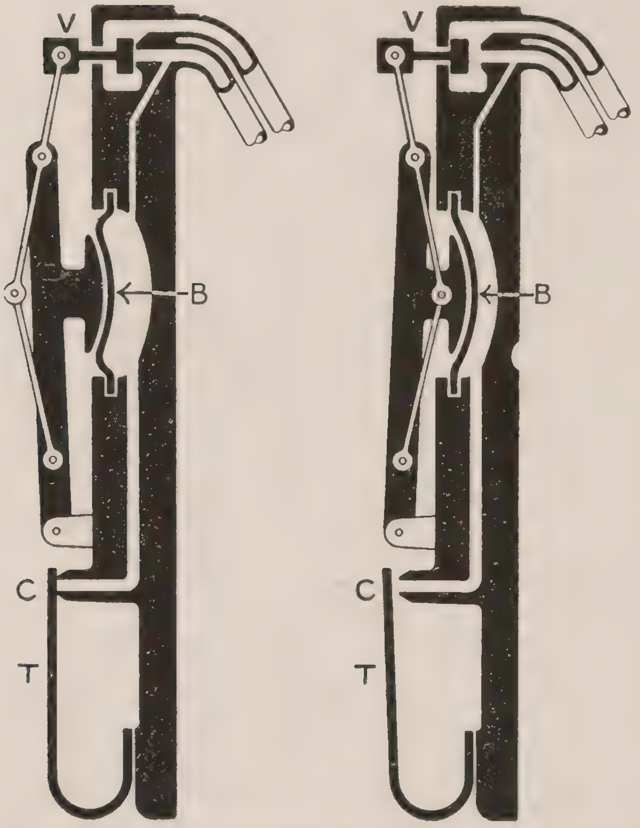


FIG. 26.—(*a*) Compressed Air Thermostat. Thermometer Bulb containing compressed air controlling mechanism. (By the courtesy of the Johnson Service Co., Ltd., Milwaukee, Wis.)

The Steam Fittings Co.'s (English) consists of a contact thermometer and electromagnetically (solenoid) controlled valve. Whenever the temperature is reached for which the contact thermometer is set a current flows (from a two-volt accumulator, or the supply main) through an electromagnet and closes the heat control valve. Immediately the temperature drops 1° to 2° the contact on the thermometer is broken, the electromagnet loses its energy, and the valve reopens. The consumption of the electromagnet when in circuit is equivalent to 200 hours' *continuous* use for one Board of Trade unit, or probably 500 or 600 working hours.

3. Wet and Dry Bulb Thermometers (Hygrometers).

In order to ascertain the relative humidity of the air, two of the thermometer tubes referred to in par. 1 are mounted together upon the same backing, but *separated* so that one cannot influence the other. The bulb of one of these thermometers is covered with a muslin cloth, which is kept



T = Compound Metal Strip.
C = Orifice.
V = Valves connected to diaphragm - operated valve.

FIG. 26.—(*b*) Compressed Air Thermostat. Details of Air Controlling Mechanism. (By the courtesy of the Johnson Service Co., Ltd., Milwaukee, Wis.)

moist by immersion in a small flask of water. The evaporation of moisture from this muslin cools the thermometer bulb so that the temperature registered by this thermometer is lower than that registered by the dry bulb thermometer. As the rate of evaporation from the wet bulb is dependent upon the relative humidity of the surrounding atmosphere, the degree of cooling, or the difference in the readings of the two thermometers, bears a definite relation to the relative humidity.

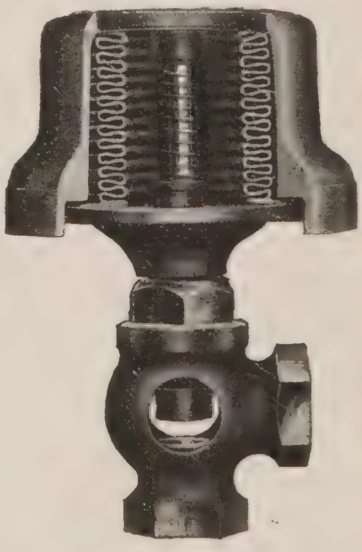


FIG. 26. — (*c*) Compressed Air Thermostat. Diaphragm-operated Control Valve. (By the courtesy of the Johnson Service Co., Ltd., Milwaukee, Wis.)

As we have standardised the room temperature as registered by the dry bulb at 60° Fahr., the calculation of the relative humidity from the wet bulb depression can be easily obtained from the Psychrometric Chart.

4. The " Kata " Thermometer.

An interesting development of the wet and dry bulb instrument, termed the " Kata " Thermometer or " Comfort Meter," is due to Dr. Leonard Hill, who, as previously stated, is of the opinion that the fundamental principle of ventilation is not one of chemical purity of the air supply but of its physical effect on the skin.

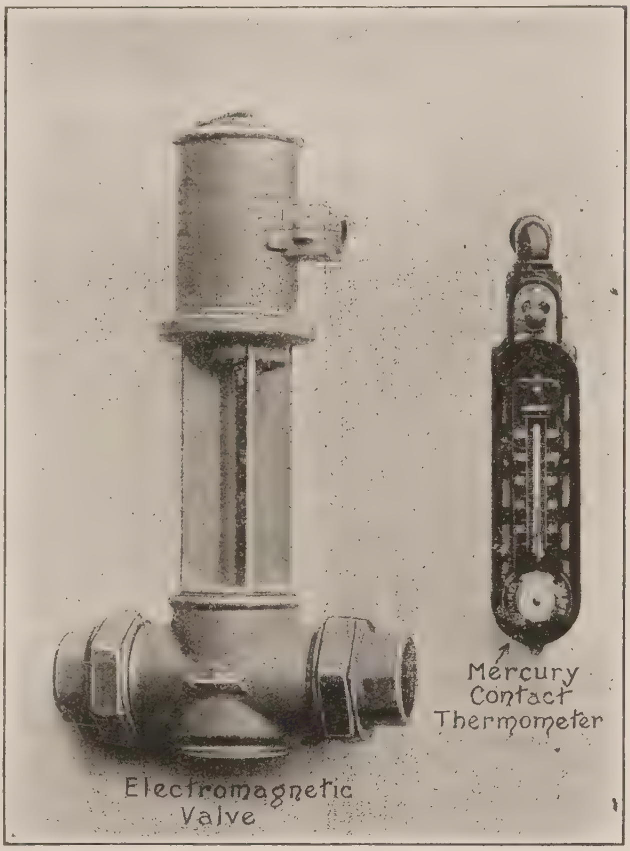


FIG. 27. — Electromagnetic Thermostat. (By the courtesy of the Steam Fittings Co., Ltd., West Drayton.)

This instrument, which consists of two large - bulbed spirit thermometers, one having the bulb fitted with a cotton finger-stall, is designed to measure *the rate of cooling of the wet and dry bulbs at or about body temperature.* The method of using is to fix the two bulbs into a clip and plunge them in water until a temperature well in excess of 110° Fahr. is registered. The instrument is then removed, suspended in the air, the dry bulb dried, the excess moisture jerked off the other bulb, and rate of cooling from 110° to 100° and thence to 90° Fahr. of each bulb registered by stop-watch.* The dry bulb loses heat by convection and by radiation ; the wet bulb also loses heat by evaporation (sweating) of the

* The same bulb may be used for both readings, the cotton finger stall being put on for the wet reading. Each "Kata" has its factor determined by which the readings are converted into micro-calories per square cm. of surface per second.

FIG. 28.—Wet and Dry Bulb Thermometer (Hygrometer). (By the courtesy of Messrs. Negretti and Zambra.)

skin, so that if the instrument is calibrated under ideal atmospheric conditions there appears to be no real difficulty in reproducing these conditions artificially in a ventilated apartment.*

5. Humidostats.

Humidostats are merely wet bulb thermostats which are installed to control automatically the wet bulb temperatures in exactly the same manner as the thermostat controls the dry bulb temperature, the difference in degrees Fahr. for which the two are set governing the relative humidity of the conditioned air.

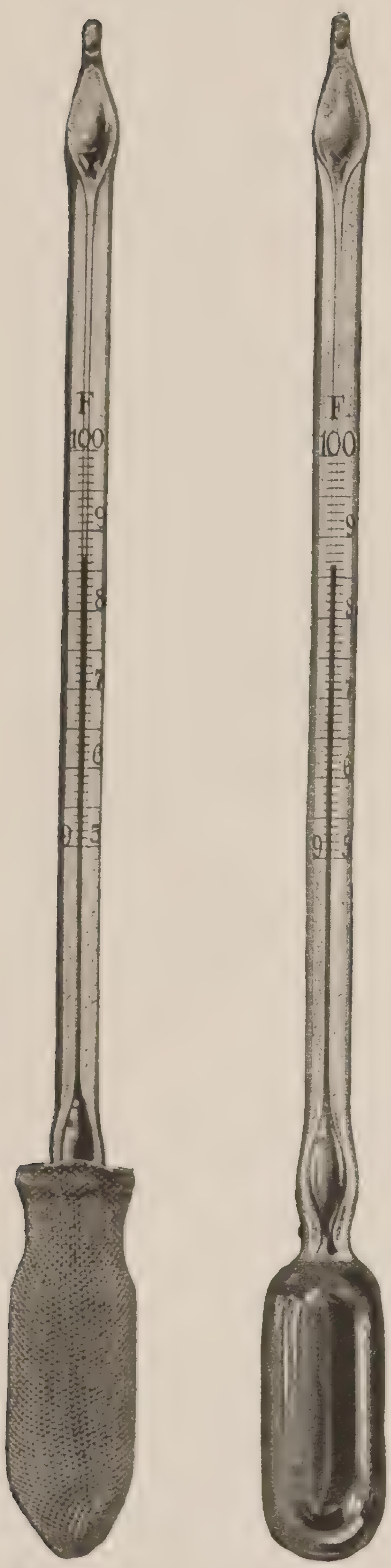
6. Polymeters.

The Polymeter, as designed by

* The readings from the dry bulb of the "Kata" give the rate of heat loss by radiation and convection; that from the wet bulb by radiation, convection and evaporation, and the difference in the two readings is obviously the rate of heat loss by evaporation alone. Dr. Hill has kindly supplied the author with the following results.

Type of Atmosphere.	Heat Loss in Micro-Calories per Square cm. of Surface Area per Second at temperature of body.		
	Radiation, Evaporation and Convection.	Radiation and Convection.	Evaporation.
Stagnant air	15·0	4·5	10·5
Bright, pleasant May day out of doors .	27·2	7·5	19·7
Fine, bracing spring day . . .	34·7	11·8	22·9
A sunny, warm day in May . . .	19·6	6·3	13·3
Indoor conditions to be produced by ventilation	18–20	7–10	10

For further details of this interesting subject, see Reports to the L.G.B. on Public Health and Medical Subjects, Series "Report on Ventilation and the Effect of Open Air and Wind on the Respiratory Metabolism, 1914," by Professor Leonard Hill, F.R.S.; for theory and use of "Kata" thermometer, see the Transactions of the Royal Society, Vol. B, 1915.



Wet Bulb. Dry Bulb.

FIG. 29.—The "Kata" Thermometer. (By the courtesy of Dr. Leonard Hill, F.R.S.)

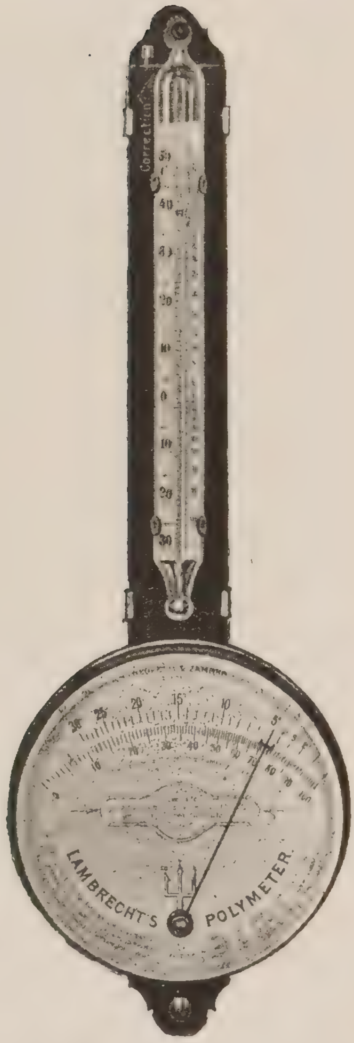


FIG. 30.—Lambrecht's Polymeter. (By the courtesy of Messrs. Negretti and Zambra.)

Lambrecht, for reading the relative humidity direct depends upon the fact that the length of a prepared human hair is affected by the hygrometric condition of the atmosphere. Obviously, therefore, it is a simple matter to suspend a small bunch of hairs from a fixed point and to measure the

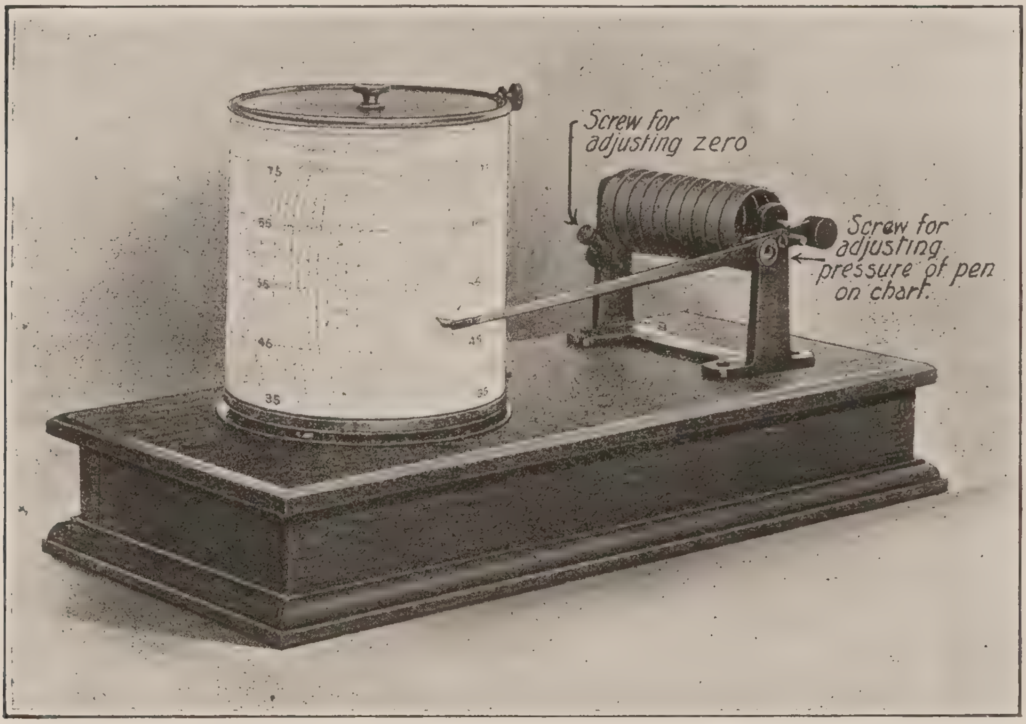


FIG. 31.—Eight-Day Thermograph.

extension or contraction in length, directly on a semi-circular scale, by means of a pivoted lever and pointer.

7. Recording Instruments.

In large ventilating installations it is convenient for the engineer to have daily or weekly records of the dry bulb temperature and relative humidity both of the conditioned and the intake air. This practice also avoids complaints by keeping the attendant up to his work, for a tell-tale mark which will lead to questions later is the inevitable result of neglect. Recording thermometers, or thermographs as they are called, are operated by the variation in length of a metal strip due to change in temperature, or vaporisation of a liquid with alteration of dry bulb

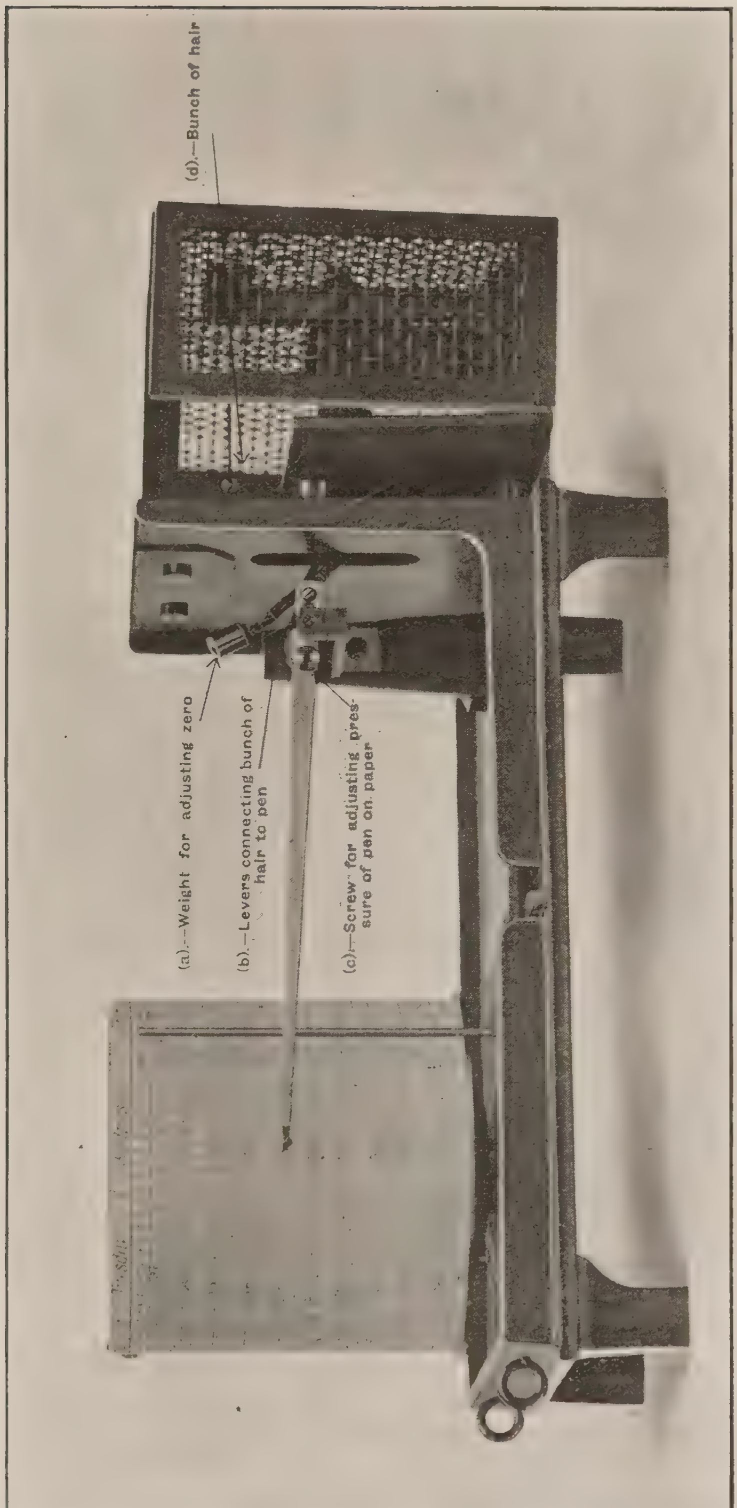


FIG. 32.—Eight-Day Hygrometer. (By the courtesy of Messrs. Negretti and Zambra.)

temperature, and self-registering hygrometers depend upon the hair principle, as in the indicating polymeter.

8. Anemometers.

The anemometer, used for measuring the velocity of air flow, consists of a light, well-balanced, and delicately con-

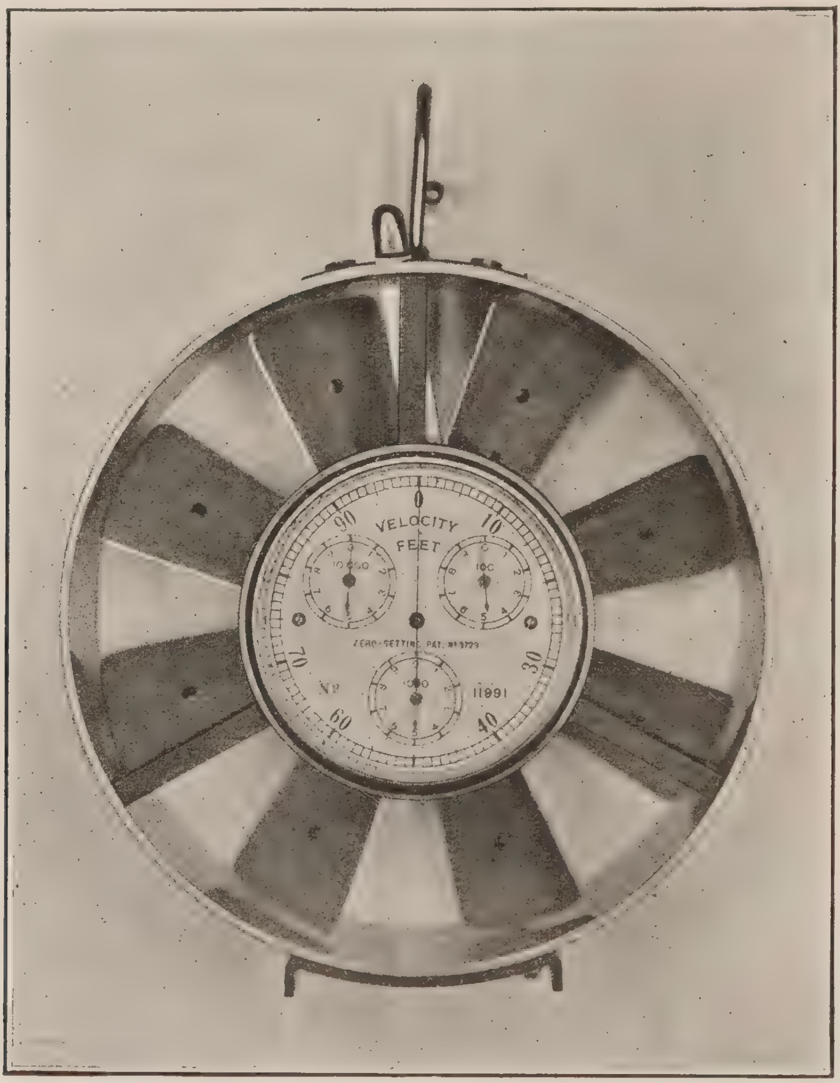


FIG. 33.—Anemometer. (By the courtesy of Messrs. Negretti and Zambra.)

structed fan wheel, whose motion is transmitted to a system of practically frictionless gearing within an attached case. Upon a dial the movement of the fan wheel is registered by hands revolving over graduated circles, and the velocity of the air in feet per minute is thus recorded.

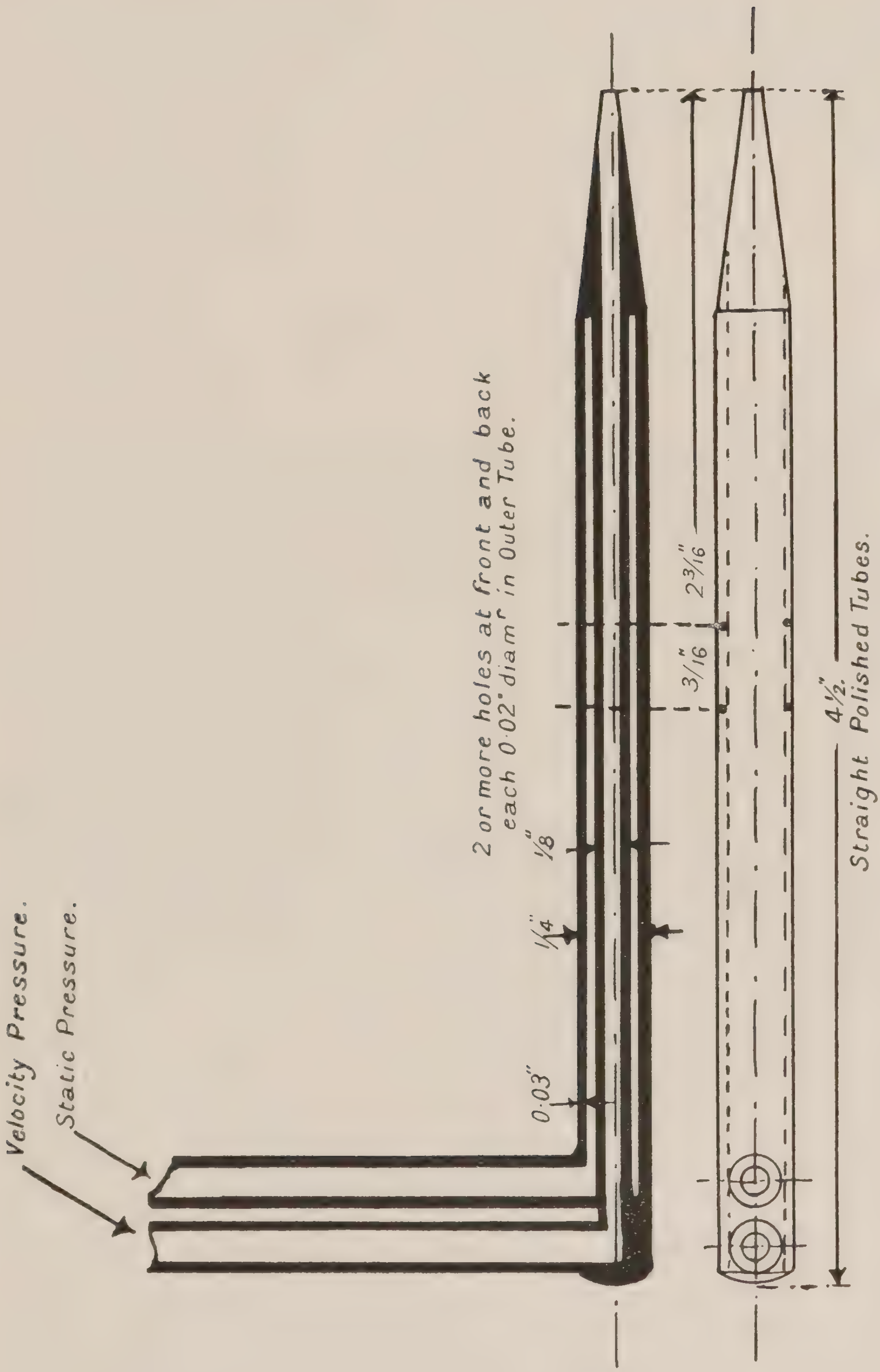


FIG. 34.—"A. B. C." Pitot Tube. (By the courtesy of the American Blower Company, Detroit, Mich.)

By moving an anemometer directly over a register or across a duct through which air is discharged, the fan wheel, being held perpendicularly to the flow of air, is revolved by the action of the moving air against its blades. The time is noted, the dial read, and the velocity of the air ascertained. Evidently the velocity thus obtained, corrected for any known error of the instrument and multiplied by the area of the passage, must give the volume. Anemometers should frequently be calibrated by the makers.

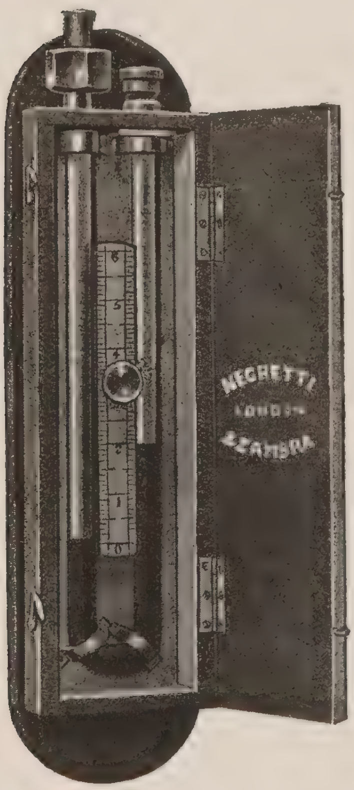


Fig. 35.—Differential Draught Gauge. (By the courtesy of Messrs. Negretti and Zambra.)

9. Pitot Tubes.

For accurate velocity readings at air velocities in excess of 1,000 feet per minute the Pitot tube gives very reliable results. It consists essentially of two parts, one of which (the inner tube) is termed the "impact nozzle," for determining the total pressure, and the other, the outer tube (pierced with small holes) for determining the static pressure.

These two tubes are connected up to the inclined draught gauge by means of small-bore rubber tube, special care being taken to have all connections tight. As shown in Chapter III., the basis for calculating velocity of air flow is similar to the formula for the velocity of bodies falling freely in space, *i.e.*,

$$V^2 = 2gh \text{ or } V = \sqrt{2gh},$$

from which at 60° Fahr. and 29·9 inch barometer,

$$V = 3950 \sqrt{h}.$$

The velocity of air through any regular duct is greater at the centre than at the sides, due to the retarding or

frictional effect of the walls. It is therefore necessary to obtain readings of velocity at five or more points, preferably ten, spaced equidistant across the diameter. Needless to say, the Pitot tube nozzle should be placed with the point facing the air current and readings taken at a place where the flow of air is as nearly uniform as possible.

10. Draught Gauges.

The pressure exerted by moving air in a duct is usually measured by the *difference* in level of a liquid in the arms

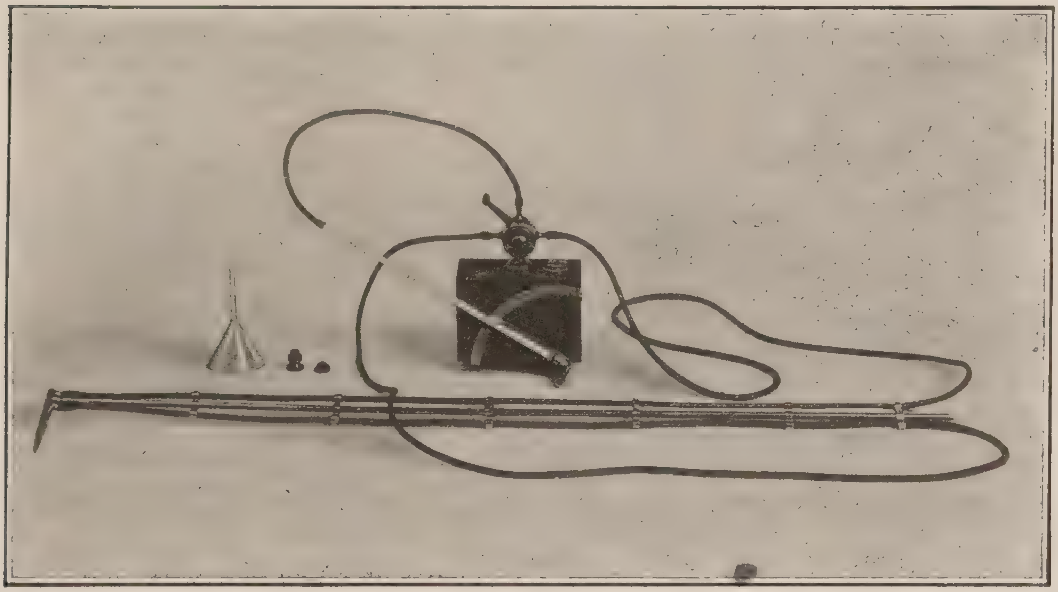


FIG. 36.—Inclined Draught Gauge and Pitot Tube. (By the courtesy of the Blackman Export Co., Ltd.)

of a U-shaped tube, having one end open to atmosphere and the other end connected to the duct or pipe in which the pressure exists.

The difference in level represents the height of a column of the liquid which will be sustained by the excess of pressure. Water is used in these gauges when employed in ventilating work.

A more accurate form is the inclined gauge used for measuring the small differences in pressure obtained in tests dealt with in par. 9. Frequently it consists of a glass tube of small bore communicating with a comparatively

large chamber forming a reservoir for the liquid and inclined at a slight angle, which in some types is adjustable. If the inclination of the tube is 1 in 10, then the pressure corresponding to $\frac{1}{10}$ inch is registered on a scale under the inclined tube by a displacement of 1 inch along the tube. Alcohol (petrol or paraffin) is used almost exclusively with this gauge for the measurement of low pressures, as it automatically keeps the bore or the tube clean, has a definite meniscus or surface, and a practically negligible capillary attraction for the glass.

11. Ammeters.

When using electric drive the ammeter connected in the motor main circuit forms a convenient method of reading directly the load on the fan. It can, if desired, be fitted with a second scale calibrated in cubic feet of air output, or for air changes for any particular installation as the result of tests.

The full-load point should be at about 50 to 60 per cent. of the scale reading in order to avoid damage to the instrument in the event of accidental overload on the motor. The moving coil type of instrument on direct current circuits has an even scale over the whole range.

CHAPTER X

GUARANTEES FOR COMPLETE INSTALLATIONS

1. Apparatus Installed.

The complete installation to be of sound material design and workmanship and to operate quietly at all duties up to specified maximum output. All parts proving defective within one year of complete delivery in working order to be replaced free of cost.

2. Fans.

The plenum and extraction fans to be capable of handling .... cubic feet of air per minute measured at 60° Fahr. and 29·92 inches barometer when all ducts, washers, heaters, registers, protective grids, and other fitments likely to cause resistance to air flow are in place.

3. Heaters.

The heater shall be of ample capacity to heat and control the humidity of the specified quantity of air from an initial condition of 30° Fahr. and 70° relative humidity to 60° Fahr. and 50° relative humidity at the plenum registers fixed in the main apartments.

4. Boilers.

The boiler to be of ample capacity to maintain continuously the specified gauge pressure (for steam) or boiler flow and return temperatures (for hot water) on the six hours' stoking basis under the specified full-load conditions, the fuel consumption not exceeding 7 lbs. for each 60,000 cubic feet of air conditioned and delivered per hour. (Fuel figured at 12,500 B.Th.U. per lb., and boiler efficiency at 64 per cent., *i.e.*, 8,000 B.Th.U. per lb. of fuel to be available in heating circuit.)

5. Motor Consumption.

The motors continuously to operate the fans and circulating pumps for six hours at the specified output, on the under-mentioned consumptions, in Board of Trade units per hour :—

Motor.	Consumption for Run.	Consumption per Hour.
Plenum fan ...	....	....
Washer circulating pump	....	....
Exhaust fan ...	....	....

6. Ducts and Outlets.

All ducts to be thoroughly air-tight, discharging air only where specified.

Velocities in mains, risers, and through registers not to exceed the limits shown on drawings.

7. Air Washer.

To remove 98 per cent. of the solid material carried by the entering air.

To remove all trace of free moisture and fog.

8. Humidity and Temperature Control.

The apparatus automatically to control the relative humidity of the conditioned air to within 1° of 60° Fahr. dry and 53° Fahr. wet bulb, provided always that the outside dry and wet bulb temperatures are less than these.

9. Cooling.

In summer when operating with re-circulated water at maximum specified duty the temperature of the conditioned air to be reduced 70 per cent. of the entering wet bulb depression.

Example 17.—If dry bulb is 82° Fahr. and wet bulb 67° Fahr., then the depression is 15°, and 70 per cent. would be 10·5; hence dry bulb would read 71·5° Fahr.

When handling cold water in summer the reduction of the outgoing difference in air and water temperatures to less than 25 per cent. of the difference in the incoming temperatures.

EXAMPLE 18.—Inlet air 82° Fahr. and water 54° Fahr. Difference — 28° Fahr. and 25 per cent. of this is 7° Fahr.

CHAPTER XI

TESTS

1. Scope.

The tests suggested in this chapter are only those required to prove for all practical purposes that the complete installation meets the specified conditions and guarantees, so that delivery can be accepted by the purchaser from the contractor after it has been proved beyond all question of doubt that the contractor has successfully selected and installed apparatus which will produce the results the purchaser desires.

2. General Inspection.

A careful and detailed inspection will be made of all registers, ducts, heaters, washers, motors, pumps, and other work carried out under the contract, to see that it has been constructed and completed in a workmanlike manner and that all débris, tools, surplus gear, etc., has been cleared, the installation being then in a thoroughly sound and usable condition.

3. Quantity of Air.

The correct measurement of the quantity of air handled is a difficult matter, and a definite understanding should be arrived at at an early date between purchaser and contractor as to the position and method of measurement.

The author is inclined to favour the Pitot tube for the main duct and anemometers for ascertaining register velocity, the total quantity being taken at a point in the duct where the air flow is free from eddies and whirls. Many readings should be taken and the average obtained.

In taking the velocity pressure by means of a Pitot tube,

the connection between the two legs of the tube and the two sides of the gauge should be the same, whether the

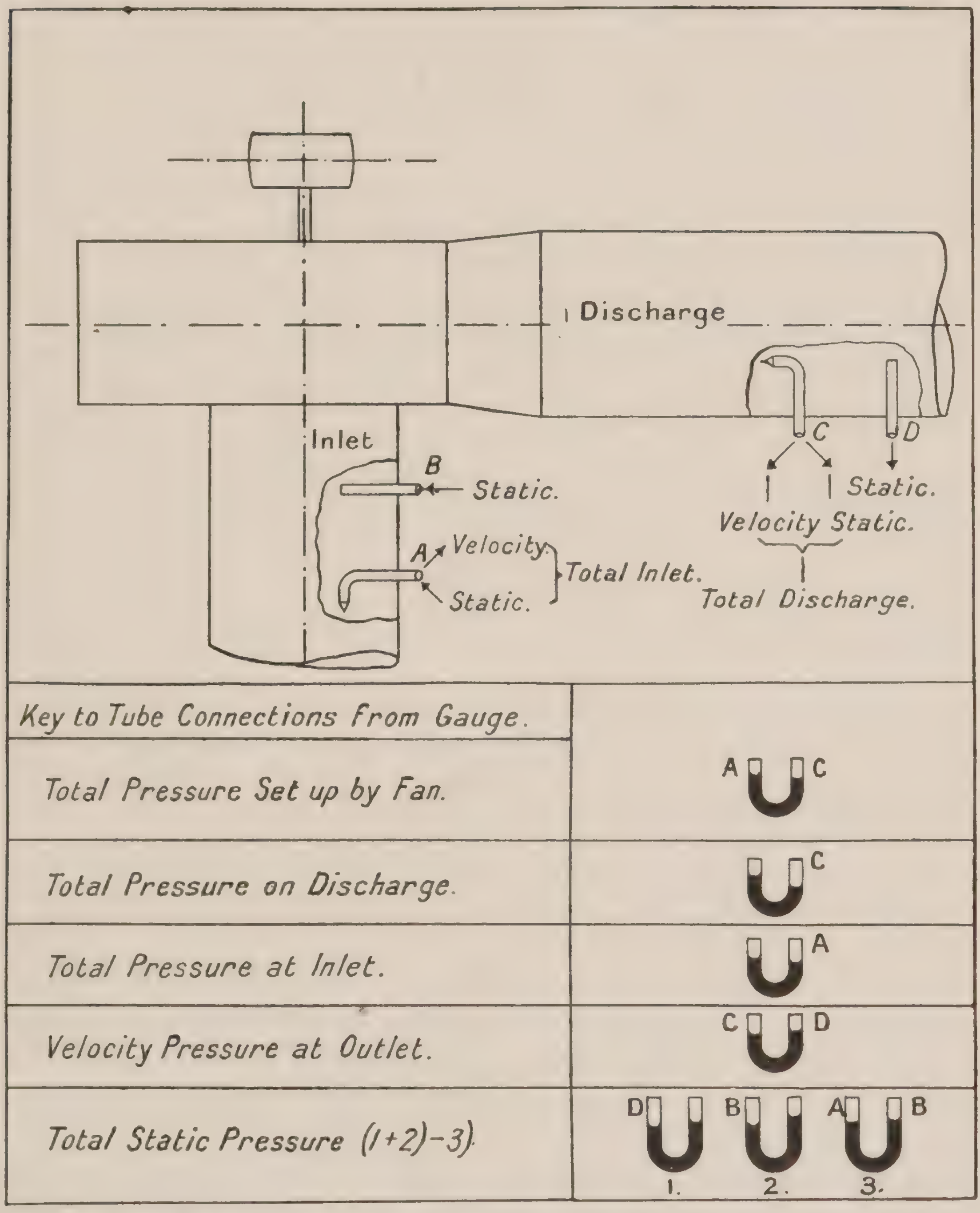


FIG. 37.—Fan Pressure Tests. (*Connect leg of Draught Gauge to branch of Pitot Tube marked with corresponding letter.*)

readings are taken in a duct either on the inlet or on the outlet side of the fan.

The total pressure set up by a centrifugal fan equals the sum of—

(*a*) The static pressure at the fan outlet.

(*b*) The velocity pressure at fan outlet.

(*c*) The static vacuum (draft or suction) or negative static pressure at the fan inlet.

(*d*) Less the velocity pressure at the fan inlet (*i.e.*, air is at rest and has to be set in motion; hence total velocity pressure is the difference of pressure at the discharge and the suction).

Since the static pressure produced by a fan equals the total pressure minus the velocity pressure at the fan outlet, it follows that

Static pressure of fan equals { The static pressure at fan outlet plus the static vacuum at fan inlet (or negative static pressure), less the velocity pressure at fan inlet.

N.B.—The static vacuum at the fan inlet minus the velocity pressure at the fan inlet is the total pressure *at that point.*

The proper arrangements of the gauges and the points at which the readings should be carefully considered.

Summarising :—

(*a*) The velocity pressure is the pressure corresponding to the average velocity over the area of either the inlet or outlet of the fan.

(*b*) The total static pressure produced by the fan is the sum of the static readings on the two sides of the fan minus the velocity pressure at the fan inlet.

(*c*) The total pressure is the sum of the velocity and static pressures.

(*d*) Quantity of air in cubic feet per minute is the product of the lineal velocity in feet per minute and the net or free area of the duct in square feet.

(*e*) In the event of electrical drive the total power input is :

(1) On continuous current circuits the product of the

volts times the amperes as shown by the meter connected at the motor.

(2) On alternating current circuits it is the product of the phase volts times the phase amperes, times the power factor of the circuit, times the number of phases connected.

(*f*) The motor losses at full load are most conveniently obtained from the makers' test sheets.

(*g*) Belt and bearing loss may be obtained by removing the runner from the shaft and taking power readings, or, if this is impossible, the loss may be assumed at from 3 to 8 per cent. of the full load of motor, depending on the tightness of the belt and whether the fan is direct-coupled or belt driven.

(*h*) The actual power consumed by the fan will then be the total watts input less the no load watts.

(*i*) The brake horse-power of the fan will be item (*e*) divided by 746.

(*j*) The total air horse-power will then be cubic feet of air per minute × 0·000157 × total pressure (inch W.G.).

(*k*) The static air horse-power will be cubic feet of air per minute × 0·000157 × static pressure (inch W.G.).

(*l*) The respective efficiencies are obtained by dividing items (*j*) or (*k*) as required by the brake horse-power input (item (*i*)).

4. Heater.

Two twenty-four hour recording dry bulb thermometers and two recording hygrometers well screened from direct draft should be placed, one in the duct leading from the fan and another in the intake duct, where it will not be affected by the radiant heat of the tempering coils, the sun, or the boiler flue.

5. Boiler.

Probably it is desirable to arrange for a twelve-hour run of the boiler, giving three hours overlap before the start

and three hours after the completion of the test. In this event the boiler is taken over at the start of the run, charged with fuel for an unattended six hours run, and the weight of fuel added at the end of the run to make up for that consumed, carefully weighed.

Any dispute as to the calorific value of the fuel can be referred to the city analyst, who for a fee of one guinea would report as to whether it could be considered an average sample or as containing an excessive amount of moisture, ash, etc. Needless to say, fuel should be reasonably dry before use to get even 8,000 B.Th.U. per lb. weight consumed out of it.

6. Motors.

Kilowatt-hour (Board of Trade unit) meters can nearly always be borrowed temporarily from the supply authority, and if any dispute as to accuracy arises a certificate can be obtained for half a guinea.

7. Air Washer.

Samples of air can be taken in a large bottle and analysed by the city analyst if considered desirable. Both samples—one in the intake duct and the other in the duct near the eliminator—should, of course, be taken at the same time in order to allow of comparison. A simpler method is to use white paper coated with sweetened condensed milk, which readily holds and shows up any particles of soot, etc.

8. Free Moisture.

A sheet of plate glass will readily indicate the presence of any free moisture as drops of water on the surface.

9. Humidity and Temperature Control.

Temperatures and also humidities are readily obtained from wet and dry bulb thermometers, or the latter from polymeters direct. Recording instruments are useful for this purpose, the type giving a twenty-four hour record being preferred.

10. Cooling.

The above remarks also apply here.

11. Loss of Draught Pressure.

The constructor and designer will naturally take steps to ascertain draught loss at various points in the air circuit by means of the Pitot tube and water or alcohol gauges used, vertical or inclined, according to the pressure to be read for future use.

Particular points of pressure drop and draught loss to be noted are :—

(1) Intake to tempering coil.
(2) Tempering coil.
(3) Washer, including scrubber and eliminator.
(4) Re-heating coils.
(5) Total draught loss on suction side, which should be equal to the sum of 1, 2, 3, and 4 with by-pass dampers both closed and open.
(6) Main ducts.
(7) Through registers.
(8) Total draught loss on discharge side (6 and 7).

In the event of the fan being fitted with a cone or diverging outlet or nozzle, the Pitot tube should be inserted at the end of the cone distant from the fan, and when there is no cone, at the fan discharge. By the former method the loss in conversion of the more or less useless high velocity pressure into useful static pressure is included, thus giving a truer indication of the real efficiency of the fan.

12. Specimen Test Sheet.

SUMMARY OF TEST TAKEN ON THE VENTILATING INSTALLATION AT

....

Designed by Messrs..

....

Job No.......

....

Details 1—8 *inclusive to be completed before sheets are issued to Test Department.*

1. Fan System No..........
2. Test in charge of.........
3. Figuring checked by.........

PLANT.

	Inlet.	Extraction.
4. *Fans.*		
(*a*) Maker		
(*b*) Type		
(*c*) Rated Duty in Cubic Feet per Minute at 60° Fahr.		
(*d*) Rated Static Pressure		
(*e*) Velocity at Discharge		
(*f*) Add for Velocity Pressure		
(*g*) Total Pressure		
(*h*) Rated Speed R.P.M.		
(*i*) Rated Power at Fan Pulley, B.H.P.		
(*j*) Serial No. of Fan		
(*k*) Drive		
(*l*) Pulley		
5. *Motors.*		
(*a*) Maker		
(*b*) Type		
(*c*) Rated duty in B.H.P.Output ...		
(*d*) Rated duty in K.W. Input		
(*e*) Full Load Amperes		
(*f*) Consumption in B.T.U. per Hour		
(*g*) Rated Speed in R.P.M.		
(*h*) Serial No. of Motor		
(*i*) Drive		
(*j*) Pulley		

6. *Air Washer.*
   (*a*) Maker ....
   (*b*) Type ....
   (*c*) Rated Duty measured at 60° Fahr. ...... cubic feet per minute.
   (*d*) Friction Loss between Inlet and Discharge ...... inches W.G.
   (*e*) Gallons of Water circulated per Hour ....
   (*f*) Pressure on Spray Nozzles ...... lbs. per sq. inch.

Test Sheet Summary at Full-rated Output of Plenum and Extraction Fans.

Time.	Motor Consumptions.						Plenum Fan.					Extraction Fan.					Air Washer.					Heater.		Boiler Temperature and Humidity.							Cooling.	
	Plenum.		Extraction.		Air Washer.		Duty in cubic ft. per min.	Speed (R.P.M.).	Pressures in Inches W.G.			Duty in cubic ft. per min.	Speed (R.P.M.).	Pressures in Inches W.G.			Water in Galls.		Evaporated in grains per cubic foot per hour.	Pressures in lbs. per sq. inch.		B.Th.U.		Gauge Pressure or Flow Temperature.	Retn. Temperature (H.W. only).	External.		Internal.		Water added (grains per cubic ft.)	Wet Bulb Depn. (° Fahr.).	% Dry Bulb Temperature Redn.
	Amps.	B.Th.U.	Amps.	B.Th.U.	Amps.	B.Th.U.			Total Static.	Velocity at Outlet.	Total Pressure.			Total Static.	Velocity at Outlet.	Total Pressure.	Pumped.	Evaporated.		Spray.	Scrubber.	Tempering Coil.	Re-heated Coil.			Dry Bulb (° Fahr.).	Relative Humidity. %	Dry Bulb (° Fahr.).	Relative Humidity. %			
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33

34. *Loss of Static Draught Pressure at*	Inches W.G.
(a) Tempering Coil	
(b) Air Washer	
(c) Re-heating Coil	

35. *Air Velocities at*	Feet per min.
(a) Main Intake	
(b) Duct to Plenum Fan	
(c) Through Free Area of Heater	
(d) Through Spray Chamber...	
(e) Through Re-heating Coil...	
(f) At Fan Discharge	
(g) Main Trunk	
(h) Branches to Risers	
(i) Risers	
(j) Through Registers	

36. *Fuel consumption for test in pounds weight* ... ...

(*g*) Pressure on Scrubbing Plate Nozzles ...... lbs. per sq. inch.
(*h*) B.H.P. of Motor ....
(*i*) Consumption per Hour ...... B. of T.U.
(*j*) Full Load Amperes ....
(*k*) Filtering Efficiency ...... per cent. of Solid Matter.
(*l*) Humidifying Efficiency ...... per cent.
(*m*) Remove all Trace of free moisture.
(*n*) Cool to ...... per cent. of the entering Wet Bulb Depression.

7. *Heater and Humidifier.*	Tempering Coil.	Re-heating Coil.
(*a*) Maker		
(*b*) Type		
(*c*) Rated Duty in B.Th.U.		
(*d*) Area in feet super.		
(*e*) B. of T.U. per foot super		
(*f*) Friction Loss in inches W.G.		
(*g*) Heating Medium per Hour (lbs. of steam at 5 lbs. gauge or gallons of hot water at mean temperature of 180° Fahr.)		

(*h*) Lbs. of 8,000 B.Th.U. Fuel per lb. to boilers at rated duty, per hour ....

8. *Velocities in the Air Circuit* (*Maximum*).	Feet per min.
(*a*) Main Intake from Street or Area	
(*b*) Duct to Plenum Apparatus	
(*c*) Through free Area of Heater	
(*d*) Through Washer	
(*e*) Fan Discharge	
(*f*) Main Distributing Trunk	
(*g*) Branches to Risers	
(*h*) Risers to Registers	
(*i*) Over Face of Register	
(*j*) Through Extraction Registers	
(*k*) Extraction Registers to Main Duct	
(*l*) Main Duct to Fan	
(*m*) Through Fan (if of Ring Type)	

CHAPTER XII

MOTOR DRIVES

1. General.

For power work in public building ventilation electric motor drive is extremely economical and convenient, but in order to obtain the best results the purchaser must exercise care in the purchase and erection of the machines.

2. Electricity Supply.

Electrical energy is supplied from street mains in four different forms—

(*a*) Continuous Current
(*b*) Single Phase }
(*c*) Two Phase } Alternating Current.
(*d*) Three Phase }

The pressure at which it is supplied (termed "voltage") varies in all four cases, the "standards" being 115, 220, 440, and 500 volts (Dublin, 346 volts).

Again, on alternating current supplies the periodicity, cycles, or the number of alternations per second (written ∼) varies, the standard being 50, but 25, $33\frac{1}{3}$, 60, and also 100 are unfortunately not unknown.

Therefore, before sending out inquiries for prices the supply authority should be requested to state—

(*a*) Type of supply to be given to premises situated in ——— Street. Continuous or Alternating ?

(*b*) Voltage of supply for motors.

(1) Up to and including five horse-power ?
(2) Above five horse-power ?

(*c*) Phases, *i.e.*, Single, Two or Three }
(*d*) Periodicity (∼) } if supply is alternating.
(*e*) Power Factor of motor at full load }

3. Types.

There are three main types of motors—

(*a*) Open type.
(*b*) Semi-enclosed.
(*c*) Totally enclosed.

Very few supply engineers will accept the open type for connection to their mains, owing to the unprotected condition of "live" metal parts, and these, therefore, are not recommended.

Semi-enclosed motors are extensively used when not exposed to wet, excessive dust, inflammable gases, etc., and should be put down whenever possible.

Enclosed motors deliver less power (30 to 50 per cent.) than semi-enclosed machines because of higher heating losses due to lack of ventilation. They have the advantage of being dust and moisture proof, and are used on direct-coupled fans and in exposed positions when trouble from these causes may be reasonably anticipated. Speeds of totally enclosed motors should be lower than those of the semi-enclosed type of equal capacity.

Motors are further divided into two main classes, depending on the type of electrical energy which they are designed to utilise—

(*a*) Continuous Current Motors.
(*b*) Alternating Current Motors.

All motors consist essentially of two parts—

—	On Continuous Current.	On Alternating Current.
(*a*) Fixed Part, termed .	Magnet case, frame or body casting.	Stator
(*b*) Moving Part, termed .	Armature	Rotor

Continuous current machines are further classified according to method of field winding adopted as follows :—

(*a*) *Series Wound*, for use when mechanically coupled to the load and when large starting torque (*i.e.*, turning

power at moment of starting) is required, as in lifts, cranes, rolling mills, etc. The speed varies with the load, and therefore series motors must never be used when the load may be entirely removed as in water pumps, belt-driven fans, etc., otherwise the speed would rise to a dangerous value.

(*b*) *Shunt Wound* machines give practically constant speed at all loads, and are therefore suitable for fan and similar work.

(*c*) *Compound Wound* machines are fitted with both series and shunt coils, thus having the combined characteristics of constant speed with good starting torque.

Alternating current machines wound for either a single, two, or three phase supply are further classified according to the method of winding the rotor :—

(*a*) *Short Circuit Rotor* or *Squirrel Cage* type, in which the rotor coils consist of copper bars connected together at the ends by metal rings. This is an extremely simple form of machine and strongly recommended for fan work up to 2 or 3 B.H.P., the only wearing parts being the bearings. There are no external cable connections to the rotor, but if the stator is switched directly on to the supply mains instead of through a transformer the momentary current will be from three to six times full-load current, the starting torque being from one to one and a half times full-load torque.

(*b*) *Wound Rotor* or *Slip-Ring* type, in which the ends of the coils on the rotor terminate in slip-rings, which connect, through carbon brushes and cables, to an external starting switch. This type is recommended for (and is usually made compulsory by supply authorities for) sizes above 2 B.H.P. There is no cable connection between the rotor and stator circuits, but by inserting additional resistance in the rotor circuit the current drawn from the mains by the *stator circuit* can be controlled as desired. In two and three phase machines with wound rotor the full-load torque can be

developed with approximately full-load current. The advantage of this type as regards starting is therefore apparent. With single-phase machines the torque corresponding to full-load current is approximately 30 to 70 per cent. full torque, depending on the periodicity.

Alternating current motors, if loaded excessively, are liable " to pull out of step " and stop. Single-phase motors on a periodicity of 50 will usually sustain an overload of about 50 to 60 per cent., and two and three phase motors about 250 per cent. under similar conditions before pulling up.

For further details as regards starting switches, etc., electrical text-books should be consulted, viz., " Alternating Current and Induction Motors " (T. Harding Churton & Co., Ltd., Leeds, 2*s*.), " Electricity for Everybody " (The Electrical Press, Ltd.), etc., etc.

4. Speeds.

High-speed motors are much cheaper than those running at lower speeds, but are liable to be more noisy. Generally speaking a maximum speed of 1,500 R.P.M. up to and including 5 B.H.P. and 1,000 R.P.M. up to 25 B.H.P. will be found satisfactory.

Variable speed, continuous current motors are very convenient for obtaining reduced outputs from fans and pumps, and are easily controlled up to 100 per cent. or over variation by means of *shunt field* regulators on machines above 1 H.P., and by series resistances for motors of $\frac{1}{2}$ H.P. and $\frac{3}{4}$ H.P. A convenient range is 750 to 1,500 for machines up to 5 H.P. and 500 to 1,000 R.P.M. for 6 to 25 H.P.

Resistances in the shunt field circuit of a continuous current motor have the effect of reducing the field strength and thus increasing the speed of the motor. This forms an economical method of speed variation within considerable range, as the field current is only a small percentage of the main current, and thus the wasted energy is in similar proportion.

Series resistances, which should not be fitted to shunt-wound motors above 0·5 B.H.P., reduce the speed by inserting resistance in the armature or main circuit, and are therefore wasteful if used for long periods.

On alternating current supplies the problem is more difficult, for it is only a year or two ago that Messrs. Parkinson, of Leeds, the General Electric Co., Ltd., of London, and one or two other makers produced the first variable speed single-phase motors on a commercial scale; but of course these are very much more expensive than the simple induction type. Normally, the speed of an ordinary alternating current (*i.e.*, induction) motor depends on the periodicity and upon the number of poles for which the motor is wound. The actual speed is a few per cent. less (probably about 3 to 7), which is called the "slip."

TABLE XXXII.—**Possible Shaft Speeds for (A.C.) Induction Motors** (Speeds in R.P.M.).

No. of Poles.	Pairs of Poles.	Frequency of Cycles (∾).					
		25	$33\frac{1}{3}$	40	50	60	100
2	1	1,500	2,000	2,400	3,000	3,600	6,000
4	2	750	1,000	1,200	1,500	1,800	3,000
6	3	500	666	800	1,000	1,200	2,000
8	4	375	500	600	750	900	1,500

5. Rating.

Fans and pumps are rated in horse-power, and motors give out horse-power but consume kilowatt-hours, Board of Trade units, or kelvins, as the same unit is variously termed, the ratio of the electrical horse-power to the kilowatt being 746 : 1,000.

For fans connected to motors by belts, etc., allow an additional 5 per cent. for belt and bearing losses when calculating power required, and specify normal full load of

TABLE XXXIII.—**Current Consumption and appropriate Cables for A.C. and D.C. Motors.**

Induction Single, Two and Three Phase and Continuous Current (Current Density approximately 1,000 amps. per sq. inch for cables).

B.H.P	Single-Phase.		Two-Phase.						Three-Phase.						Continuous Current.							
	200 Volts.		200 Volts.		400 Volts.		200 Volts.		346 Volts.		415 Volts.		500 Volts.		100 Volts.		220 Volts.		440 Volts.		500 Volts.	
	Amps.	Cable.	Amps.	Cable.	Amps.	Cable.	Amps.	Cable.	Amps.	Cable.	Amps.	Cable	Amps.	Cable.	Amps.	Cable.	Amps.	Cable.	Amps.	Cable.	Amps.	Cable.
1	7·2	7/20	3·0	3/20	1·3	3/22	3·5	7/22	2·0	3/20	1·7	3/22	1·4	3/22	10	7/18	4·5	3/22	2·3	3/20	2·0	3/20
3	18	7/16	8·5	7/18	3·7	7/22	10	7/18	5·7	7/20	4·8	7/20	4·0	7/22	30	19/18	14	7/16	6·9	7/20	6·0	7/20
5	27	19/18	13	7/18	5·7	7/20	14	7/18	8·2	7/18	7·0	7/20	5·7	7/20	50	19/16	22	7/16	11	7/18	10	7/18
7½	40	19/17	19	7/16	8·5	7/18	22	7/16	13	7/18	11	7/18	8·5	7/18	75	19/14	34	19/18	17	7/16	15	7/16
10	51	19/16	24	7/16	11	7/18	28	19/18	16	7/16	14	7/16	11	7/18	100	37/16	46	19/17	23	7/16	20	7/16
15	77	19/14	35	19/18	16	7/16	40	19/17	23	7/16	19	7/16	16	7/16	150	37/14	68	19/14	34	19/18	30	19/18
20	102	37/16	47	19/17	22	7/16	55	19/16	32	19/18	26	19/18	22	7/16	200	—	90	19/14	45	19/17	40	19/17
30	145	37/14	70	19/14	32	19/18	81	19/14	47	19/17	39	19/17	33	19/18	300	—	136	37/14	68	19/14	60	19/16
40	204	—	94	19/14	43	19/17	110	37/16	63	19/16	52	19/16	44	19/17	400	—	182	—	91	19/14	80	19/14
50	236	—	115	37/16	52	19/16	133	37/14	77	19/14	64	19/14	53	19/16	500	—	228	—	114	37/16	100	37/16

In induction motors having wound rotors the rotor and stator windings are entirely separate, and before the wiring between the rotor and starting switch is commenced the motor makers should be requested to specify the maximum rotor currents. Similarly for continuous current shunt or compound wound variable speed machines the maximum value of the shunt field current should be obtained and the cable specified accordingly.

the motor at 20 per cent. in excess of the figure so obtained. For fans direct coupled to motors, the power of the motor should also be 20 to 25 per cent. in excess of the actual B.H.P. required by the fan when delivering the specified volume against the estimated resistance or water gauge of the air circuit. This apparently large margin of power is advisable because of the fact that the power required by the fan increases as the cube of the speed, as shown in Table XXXIV., and also to the difficulty of exactly determining the exact W.G. resistance of the air circuit, as if this has been under-estimated the fan will have to run faster to deliver the specified volume; if it has been over-estimated the fan will deliver more than the specified volume "unless the speed is reduced" with consequent increase of load upon the motor. In the case of alternating current motors which run at fixed speed, it is especially essential to have this margin of power in the motor.

TABLE XXXIV.—**Horse Power Required by Centrifugal Cased Fan at Various Speeds.**

No. 13 Fan: 1 inch Static Pressure (Constant).

Revs. per minute .	204	212	226	250
Horse-power required	14·1	17·7	24·4	38·6
Vol. cub. per minute .	34,800	41,800	51,300	63,500

There are two principal methods of rating motors for temperature rise—

(*a*) Crane rated.

(*b*) Continuous rated.

Motors used for crane, lift, tramcar, and similar work are stopped and started continuously, so that they do not develop full power and consume and waste energy con

tinuously; thus the heating effect is not so great as they have time to cool down to an appreciable extent between times.

On the other hand, motors which run continuously (*i.e.*, "continuous rated") attain to their maximum temperature in about four hours, after which period the heat is dissipated as quickly as it is produced, resulting in a steady temperature. This is the type to be used for all ventilating work when fans, pumps, etc., run continuously.

6. Starting up.

Where 100 per cent. speed regulation is adopted, no difficulty will be experienced in starting up the motor when coupled direct or by belt to the fan or pump. In cases where this is impossible, and especially on alternating current single-phase systems when the starting current is limited to the full-load current, it will be necessary to fit dampers to close entirely the discharge of the centrifugal fan, etc., to by-pass the air on ring-type fans, and to fit fast-and-loose pulleys, including belt-shifting devices and double-width motor pulleys, to all but the smallest sizes of fans and pumps.

7. Control Panels.

All starting panels should be provided with ammeters which can conveniently be marked with the full half and quarter load readings in red. Iron-clad double-pole (triple-pole on three-phase and possibly four-pole on two-phase supplies) starting switches fitted with overload and no voltage releases, and preferably interconnected with the field regulators, so that motors cannot be started up on a weak field (*i.e.*, at full speed) should also be included. Quarter-inch plate iron mounted on 2-inch by $\frac{3}{16}$ inch or $\frac{1}{4}$-inch strip forms a convenient support for this gear.

8. Drives and Belts.

The distance between the centre of the motor shaft and the fan or pump shaft is usually rather small, resulting in

tight belts and excessive loads. Another common fault is that of excessive pulley ratios, which should not exceed 6 : 1.

For a 5 B.H.P. machine and a 4·1 pulley ratio 8 feet centres should be considered a minimum.

For a 10 B.H.P. machine and 4·1 pulley ratio, 10 feet centres should be considered a minimum (preferably 12 feet).

For a 12 B.H.P. machine and 4·1 pulley ratio, 12 feet centres should be considered a minimum.

Belt drives, which are unusual for powers below 3 B.H.P., are another important detail, and in this case it is desirable to err on the side of excessive width, which will result in increased life to both belt and bearings.

Fan makers frequently fit pulleys the diameter of which is on the small side, and therefore it is desirable to take the motor pulley as the basis for figuring.

1,500 to 3,000 feet per minute is a very general speed for single-width belts (maximum 4,000 feet per minute), such as are used for transmitting moderate powers (say up to 40 B.H.P.), and a ratio of pulley diameter to belt width 1·5 or 2 to 1 gives good results.

A convenient formula, believed to be due to Berry, states :—"One watt of electrical energy is transmitted per foot per minute of belt speed per inch of width of single belt."

EXAMPLE 19.—Thus, taking a 25 B.H.P. motor running at 750 R.P.M. and fitted with a 12 inch by 7 inch pulley we have—

(*a*) Belt speed, $\frac{750 \times 12 \times 3\frac{1}{7}\ (\pi)}{12} = 2{,}350$ feet per minute.

(*b*) Watts output (746 watts = 1 B.H.P.), $25 \times 746 = 18{,}700$

(*c*) Belt width, $\frac{18{,}700}{2{,}350} = 7{\cdot}92$, or, say, 8 inches.

(*d*) Ratio of pulley diameter to belt width = 12 : 8, *i.e.*, 1·5 : 1.

TABLE XXXV.—**Horse-Power Transmitted by Single Leather Belts.**

Width of Belt (Inches).	Speed of Belt in feet per minute (*i.e.*, diameter of pulley in feet × π × R.P.M.).								
	1,000	1,250	1,500	1,750	2,000	2,250	2,500	3,000	3,500
1	1·3	1·6	2·0	2·3	2·6	3·0	3·3	4·0	4·7
2	2·6	3·3	4·0	4·6	5·3	6·0	6·7	8·0	9·4
3	4·0	5·0	6·0	7·0	8·0	9·0	10	12	14
4	5·3	6·6	8·0	9·4	10	12	13	16	18
5	6·7	8·3	10·0	11	13	15	16	20	23
6	8·0	10	12	14	16	18	19	24	28
7	9·3	11	14	16	18	21	23	28	32
8	10	13	16	18	20	24	26	32	37
9	12	15	18	21	24	27	29	36	42
10	13	16	20	23	26	30	33	40	47
11	14	18	22	25	28	33	36	44	51
12	16	20	24	27	32	36	39	48	56

For light double belts the power transmitted can be increased by 25 per cent. on above figures.

For heavy double belts the power transmitted can be increased by 50 per cent. on above figures.

Make the pulling half of the belt (the tight side) the lower one in order to increase the arc of contact between belt and pulley (*i.e.*, slack side on top).

9. Bearings.

Ring lubricated bearings certainly tend to quiet running, but Stauffer grease lubricators give excellent results on small ring-type fans fixed in inaccessible positions.

Ball and roller bearings form a debatable subject, but the author, having given them a good trial, has returned to the older ring lubricated type. Theoretically the ball bearings are excellent, the chief troubles experienced apparently being due to bad fitting and the selection of unsuitable types by the motor manufacturers for the work in hand. Also of course the noise of the balls revolving in the race add to rather than reduce the total noise made by the apparatus.

10. Tender Detail Sheets.

The present writer has found that a standard detail sheet is extremely useful when purchasing machinery, and especially is this so in the case of electric motors, in which there are so many alternative details. Practically the whole of the work of filling them in can be delegated to juniors and duplicates can be typed as required, thus leaving the senior staff free to tackle new problems. It is of course a simple matter to draft similar forms for fans, pumps, etc., if the business demands it. See Chapter XIV.

CHAPTER XIII

SUGGESTED SCHEDULES FOR DESIGNING

Complete Installation

Introduction.

An increase of paper work is easily seen and severely criticised by many otherwise keen business men, not on the merits and demerits of the system, but merely on the grounds that it is system. Frequently also the small-minded technical man thinks that by raising petty objections and refusing to systematise his work he renders himself indispensable—he certainly does so long as such an absurd policy is allowed to retard the output of the firm, which is just about as long as it takes to get a better man who can delegate the routine work and get busy with bigger things.

However, economy lies further along the line, and if the value of the introduction of routine forms and method is tested by the results enumerated below, a moderate system will be found amply to justify itself :—

(*a*) Increased efficiency by reduction or practical elimination of mistakes and omissions in calculations.

(*b*) Increased turnover of capital by reduction of contract time, due to possibility of routine work being handled by juniors when once the main outline of the scheme has been settled by a responsible engineer.

(*c*) Decreased friction, and hence wasted energy, between office and erecting staff by reason of delivery of exactly the plant required and the arrival of necessary detailed information.

(*d*) Increased facility with which a job can be picked up at any point by any member of the technical staff in the event of illness, etc., of the engineer handling it.

The point when system should be stopped is when there

is no advantage to be gained in *any* direction in having all the requisite information in just the same style every time.

The schedules following do not pretend to be full and complete, for that would be impossible when every engineer has his own particular methods of arriving at the desired results. They are put forward as a framework to be completed by the reader who may care to give up the haphazard methods of figuring on scraps of paper upon no predetermined method and systematise his work so that it can be rapidly checked and is available for reference years afterwards if extensions or alterations are required.

The present writer, who has used similar schedules for many years now, finds it convenient to retain different coloured files for electrical power, lighting, ventilating, heating, elevators, etc., work, securing the schedules dealing with the particular section in the appropriate file.

Designing Schedule No. 1.

VITIATION AND TOTAL AIR SUPPLY.

1. Job No.........
2. Fan System No.........
3. Apartment to be ventilated numbered on plan.........
4. External Sources of Heat :—
    (*a*) Seats or persons ....=......... B.Th.U.
    (*b*) Gas jets or cookers ....=......... B.Th.U.
    (*c*) Electric Arc Lamps or resistances ....=......... B.Th.U.
    (*d*) Steam pipes ....=......... B.Th.U.
    (*e*) Other sources of heat......=......... B.Th.U.

    Gross Total B.Th.U. .... B.Th.U.
    Less any probable losses by transmission, etc. .... B.Th.U.

    Nett total B.Th.U. to be removed per hour .... B.Th.U.

5. Temperature rise allowable .... degs. Fahr.
6. Total air required to hold temperature increase at this limit .... cubic feet per hour.
7. Ditto ditto .... cubic feet per minute.

8. (*a*) Quantity to be handled by plenum fan .... cubic feet per minute.
   (*b*) Ditto extraction fan .... cubic feet per minute.
9. Cubical contents of room .... cubic feet.
10. Changes per hour of air contents.........

Calculated by.............. Date.........
Checked by.............. Date.........

Designing Schedule No. 2.

FANS AND MOTORS.

General Details :—

1. Job No..........
2. Fan System No..........
3. System of Ventilation Adopted.........
4. Ventilated Apartment to be used for.........
5. Maximum height in fan room......ft.......ins.

Fan Details :—

6. Type.........
7. Capacity......cubic feet per minute.
8. Total static water gauge......inch (maintained resistance).
9. Velocity through discharge......feet per minute.
10. Equivalent velocity pressure......inches.
11. Total pressure to be set up by fan......inches W.G.
12. Speed......R.P.M.
13. Maker.........
14. Size.........
15. Direction of discharge.........
16. Handed.........
17. Housing.........
18. Inlet (single or double).........
19. Drive.........
20. Working position of shaft.........
21. Base on floor, ceiling or wall.........
22. Pulley :—
    (*a*) Width......inches.
    (*b*) Diameter......inches.
    (*c*) Surface......inches.
    (*d*) Number......inches.
23. Belt striking gear required ?.........
24. Type of bearings.........

25. Power at fan shaft......B.H.P.
26. Add 10 per cent. for belt losses......B.H.P.
27. Total power for fan......B.H.P.
28. Maximum height of fan......ft.......ins.
29. Ditto width ditto ......ft.......ins.
30. Ditto length ditto ......ft.......ins.
31. Total weight of fan......lbs.
32. Type of belt.........
33. Speed......feet per minute.
34. Width......inches.
35. Maximum power transmitted at that speed......B.H.P.

Motor Details :—

36. Maximum power required by fan (line 27)......B.H.P.
37. Add margin of 20 per cent.......B.H.P.
38. Total output of motor......B.H.P.
39. Electricity supply pressure and system.........
40. Type.........
41. Winding (shunt, squirrel cage, etc.).........
42. Speed :
    (a) Maximum......R.P.M.
    (b) Minimum......R.P.M.
43. Rotation.........
44. Drive.........
45. Working position of shaft.........
46. Position of magnet feet.........
47. Pulley :
    (a) Diam. ......inches.
    (b) Width ......inches.
    (c) Surface......inches.
48. Length of drive (pulley centres).........
49. Ratio in diam. of fan and motor pulleys.........
50. Assumed efficiency of motor at full load......per cent.
51. Maximum input to motor......kilowatts.
52. Maximum current per main cable......amps.
53. Appropriate size of cable.........
54. Maximum current for shunt or rotor cable......amps.
55. Appropriate size of cable.........
56. Ammeter scaled to read 0......amps., with open scale at...... amps.
57. Maximum height......ft.......ins.
    Ditto width......ft.......ins. (right angles to shaft).
    Ditto length......ft.......ins.
58. Gross weight will be......lbs.

Calculated by.............. Date.........
Checked by.............. Date.........

Designing Schedule No. 3.

AIR WASHING PLANT.

General Details :—

1. Job No.........
2. Fan System No.........
3. Apartment to be ventilated.........
4. Maximum height of room......ft.......ins.
5. Air to be handled......cubic feet per minute.
6. Maximum allowable pressure drop......inches water.
7. Maximum water to be evaporated per cubic foot of air...... grains.
8. Maximum total quantity of water to be evaporated per hour ......gallons.

Washer Details :—

9. Maximum velocity of air through spray chamber......feet per minute.
10. Cross sectional area required......sq. feet.
11. Height selected......ft.......ins.
12. Corresponding width......ft.......ins.
13. Overall length of air washer......ft.......ins.
14. Casing required.........
15. Connected to at :
    (*a*) Inlet end.........
    (*b*) Discharge end.........
16. Observation doors at......side looking in direction of air flow.
17. Size of largest part to pass through opening of......ft.......ins.
18. Size of pipe for ball valve make up......inches diam.
19. Size of drain connection......inches diam.
20. Perforated distribution plate required at entrance ?.........
21. Capacity of settling tank......gallons.
22. Total weight of washer with settling tank full of water, but excluding pump......lbs.

Pump :—

23. Type.........
24. Pressure head varied, how ?.........
25. Normal pressure head......lbs. per sq. inch.
26. Capacity......gallons per hour.
27. Diam. of suction pipe......inches.
28. Diam. of discharge pipe......inches.
29. Pump flooded when tank full ?.........
30. Foot valve and priming cock required ?.........

Motor Details :—

31. Maximum power required by pump......B.H.P.
32. Add margin of 20 per cent.......B.H.P.

33. Total output of motor......B.H.P.
34. Electricity supply system and pressure.........
35. Type.........
36. Winding (shunt, compound, squirrel cage, etc.).........
37. Speed :
    (*a*) Maximum......R.P.M.
    (*b*) Minimum......R.P.M.
38. Rotation.........
39. Working position of shaft.........
40. Position of magnet feet.........
41. Type of coupling.........
42. Assumed efficiency of motor at full load......per cent.
43. Maximum input to motor......kilowatts.
44. Maximum current per main cable......amps.
45. Appropriate cable size.........
46. Maximum current per shunt or rotor cable......amps.
47. Appropriate cable size.........
48. Ammeter scaled to read 0......amps., with open scale at...... amps.
49. Maximum height......ft.......ins.
    Ditto width......ft.......ins. (right angles to shaft).
    Ditto length......ft.......ins.
50. Gross weight of pump, motor, contained water, etc.......lbs.

Calculated by.............. Date.........
Checked by.............. Date.........

Designing Schedule No. 4.

PLENUM HEATER.

General Details :—

1. Job No.........
2. Fan System No.........
3. Maximum height of room......ft.......ins.
4. Air to be handled......cubic feet per minute.
5. Water to be evaporated per cubic foot......grains.
6. Total range of temperature......° Fahr.
7. Heating medium adopted.........
8. Average temperature of hot water or pressure of steam.........
9. Temperature increase of tempering coil :
    (*a*) Temperature required in spray chamber for humidification......° Fahr.
    (*b*) Initial temperature......° Fahr.
    (*c*) Increase required......° Fahr.
    (*d*) Add $8\frac{1}{2}°$ Fahr. for each grain of water evaporated per cubic foot (line 5)......° Fahr.
    (*e*) Total temperature increase required for tempering coil......° Fahr.

10. Temperature increase of re-heating coil :
    (*a*) Final temperature......° Fahr.
    (*b*) Temperature in spray chamber (9 (*a*))......° Fahr.
    (*c*) Increase required......° Fahr.
11. Heat units from tempering coil :
    (*a*) $\frac{\text{Cub. ft. of air per hour} \times \text{line 9 } (c)}{53}$ = air tempering.
    (*b*) $\frac{\text{Cub. ft. of air per hour} \times \text{line 9 } (d)}{53}$ = evaporation of water.

    Total heat from tempering coil if both are combined in one heater :
    B.Th.U. for air tempering.........
    B.Th.U. for evaporation of water.........
    Total B.Th.U. from tempering coil.........
12. Heat units from re-heating coil :
    $\frac{\text{Cub. ft. of air per hour} \times \text{line 10 } (c)}{53}$ = re-heating.
13. Maximum velocity of air through heater......feet per minute.
14. Cross-sectional *free* area required......feet super.
15. Type of heater selected.........
16. Ratio of *free* to gross area......per cent.
17. Gross cross-sectional area required......feet super.
18. Add $12\frac{1}{2}$ per cent. for by-pass dampers.........
19. Total gross cross-sectional area required for heater.........
20. Maximum height available......ft.......ins.
21. Corresponding width of heater......ft.......ins. (at right angles to air flow).

	Tempering.	Re-heating.
22. Number of sections at the rated percentage of free area per section required for the *free* area (line 14)		
23. Number of tiers......each		
24. Total number of sections required in each heater		
25. Number of stacks deep in direction of air flow		
26. Number of sections per stack ...		
27. Height of sections		
28. Width of sections		
29. Nipple centres		
30. Total length of stack		
31. Tapped for pipe diameter		

LOSS OF PRESSURE IN DUCT SYSTEM.

Job No. .... Fan System No. .... Maximum Velocity of Air .... feet per minute.
Maximum W.G. resistance on ducts desired .... inches.

Branch Section No.	Air Supply (Cubic Feet per Minute).	Material of which Duct constructed.	Cross-sectional Area (Sq. Feet).	Velocity of Air (Feet per Minute).	Net Length of Duct (Feet).	Add Equivalent Lengths for Bends, etc. (Feet).	Total Air Length (Feet).	Friction Loss (Inches of Water).

It will be found convenient to draw an isometric plan of duct work, divide up into sections, giving each section a distinguishing letter, and using a rubber stamp for details required against each section, thus :—

Quantity of air (cub. ft. per min.)	......	Add for bends, etc. (ft.)	......
Velocity (ft. per min.)	......	Equivalent length (ft.)	......
Friction loss per 10 ft. run (ins.)	......	Total friction loss (ins.)	......
Net length (ft.)	......	Material	......

	Tempering.	Re-heating.
32. Maximum B.Th.U. through this diameter pipe		
33. Area of sections in feet super. ...		
34. Area of heater in feet super. ...		
35. Efficiency of surface in B.Th.U. per foot super.		
36. Air frictional loss in inches of W.G.		
37. Quantity of heating medium required for specified duty (lbs. of steam or gallons of hot water per hour)		

Arrangements and connections of pipework must be designed to balance loss of pressure of heating medium in passage through heater to head available for that purpose.

CHAPTER XIV

PURCHASE SPECIFICATIONS

Introduction.

In modern engineering business management slipshod methods have no place, and the man who orders "one fan (or motor) as before," leading to endless correspondence and ultimate ill-feeling, should be given marching orders at once before he has an opportunity to do further damage.

Manufacturers have standardised apparatus to a certain extent, and in dealing with well-known firms it will probably be the wiser course for those not specialising in the various branches of engineering allied with the ventilating industry to purchase on trade descriptions rather than on detailed specifications. For those who are qualified to use the fuller form, drafts are included for ducts, fans, wet-air filters, and continuous-current motors.

It is suggested that these trade descriptions or purchase specifications should be standardised and numbered, so that the purchase order may simply contain the reference "as per attached specification," together with price, terms, etc., etc.

It was probably an American who remarked, "Why make the *same* mistakes twice when there are so many others to select from?" but this is the inevitable result if some systematic course for purchasing machinery, such as indicated in the following pages, is not adopted.

PART I.

Trade Descriptions.

Trade Description No. 1.

CENTRIFUGAL CASED FAN.

(Silent Running, Essential.)

[*Strike out the words in italics which do not apply.*]

Job No.

1. To Messrs. ....
....
....

2. Delivery required to site at...........carriage paid by .../.../19......certain/*and erected complete, ready for beneficial use* by.../.../19.........*certain.*

3. **Output:**

(*a*) ......cubic feet of air per minute measured at 60° Fahr.

(*b*) Against a maintained (static) resistance of......inches water col.

(*c*) Duct Velocity at Fan......cub. ft./min.

4. **Type:**

(*a*) *Single/Double*/Inlet.

(*b*) *Top/Bottom/Horizontal/Vertical Up/Vertical Down* discharge.

(*c*) *Pulley/Motor*/to be on/*Right/Left*/hand when looking into discharge.

(*d*) *Full/Half*/housing.

(*e*) *Single/Double*/width (latter for limited head room and extraction purposes only).

5. **Drive:**

Belt / Chain / Direct / driven by / *Direct / Alternating / Current Motor/Shafting/Steam Engine.*

(N.B.—If direct coupled to motor, use appropriate Trade Description Sheet.)

6. **Working Position:**

(*a*) Shaft/*Horizontal/Vertical.*

(*b*) Base on/*Floor/Ceiling/Wall.*

7. **Pulley:**

Single width/Fast and loose pulleys with belt striking gear, convex, of......″ diam. × ......″ width, having crowning of ......″.

8. **Bearings:**

Ring lubricated/Stauffer grease cup/Ball/Roller/bearings.

9. **Size of Largest Part:**

To pass through opening of......ft.......ins. by......ft. ......ins.

10. **Drawings Required with Tender:**

(*a*) Outline drawing of fan as erected, figured in with overall dimensions.

(*b*) Pulley centres.

(*c*) ....

11. **Technical Information** (to be filled in by Tenderer) :

(*a*) *Performance Details :*

—	Maker's Fan No.	Outlet Velocity in Ft./Min.	Add for Total Pressure (Ins. of W.G.).	R.P.M.	Diam. of Wheel.	Peri-pheral Speed of Ditto (Ft./Min.)	H.P. at Pulley.	Static Air H.P.	Static Mechani-cal Effi-ciency.
(*a*)									
(*b*)									
(*c*)									

(*b*) *Gross Weight* will be :

	(*a*)	(*b*)	(*c*)
Maker's Fan No.			
Lbs.			

....

....

....

....

....

....

Signed...............

Dated.........

Trade Description No. 2.

RING TYPE FAN.

(Silent Running, Essential.)

[*Strike out the words in italics which do not apply.*]

Job No.

1. Inquiry to Messrs.......................................
....
....

2. Delivery required to site at............carriage paid by .../.../19......certain/*and erected complete, ready for beneficial use* by.../.../19.........*certain.*

3. **Output:**
......cubic feet of air per minute on free intake and discharge, measured at 60° Fahr.

4. **Type:**
*Pulley/Motor/*to be on/*suction/discharge/*side of fan.

5. **Drive:**
*Belt/Chain/Direct/*driven by a/*Direct/Alternating/Current Motor/Shafting/Steam Engine.*
(N.B.—If direct coupled to motor, use appropriate Trade Description Sheet.)

6. **Working Position:**
Shaft/*horizontal/vertical.*

7. **Pulley:**
Single width/*fast and loose pulleys with belt striking gear/* convex, of......″ diam. by......″ width, having crowning of ......ins.

8. **Bearings:**
*Ring lubricated/Stauffer grease cups/Ball/Roller/*bearings.

9. **Drawings Required with Tender:**
(*a*) Outline drawing of fan as erected, figured in with overall dimensions.
(*b*) Pulley Centres.........
(*c*) ....

10. **Technical Information** (to be filled in by Tenderer) :

(*a*) *Performance Details :*

—	Maker's Fan No.	Air Velocity in Ft./Min. through Wheel.	Equivalent Velocity Pressure (Ins. W.G.).	R.P.M.	Diam. of Wheel (Inches).	Peripheral Speed of Ditto (Ft./Min.)	H.P. at Pulley.	Total Air H.P.	Total Mechanical Efficiency.
(*a*)									
(*b*)									
(*c*)									

(*b*) *Gross weight* will be (exclusive of motor) :

	(*a*)	(*b*)	(*c*)
Maker's Fan No.			
Lbs.			

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....

....

....

....

Signed...............

Dated.........

Trade Description No. 3.

WET AIR FILTER.

(Silent Running, Essential.)

[*Strike out words in italics which do not apply.*]

Job No.

1. Inquiry to Messrs...
....
....

2. Delivery required to site at...........carriage paid by......
.../.../19.........certain/*and erected complete, ready for beneficial use by.../.../19.........certain.*

3. **Capacity:**
To handle......cubic feet of air per min. measured at 60° Fahr.

4. **Type:**
Spray type for air washing and humidifying.

5. **Casing:**
Full casing/Make good to brickwork shown on Drawing No.......attached.

6. **Pump:**
Centrifugal type/*Variable/Constant Speed*/direct coupled to an electric motor.
(N.B.—Use appropriate Trade Description Sheet.)

7. **Pump Bearings:**
Ring lubricated/Stauffer grease cups/Ball/Roller/bearings.

8. **Size of Largest Part:**
To pass through opening of......ft.......ins. by......ft. ......ins.

9. **Maximum Height of Washer:**
......ft.......ins. (Depending on available head room. Velocity of air through filter is usually 500 ft./min., so that height may be reduced at expense of width but preferable square in cross-section.)

10. **Drawings Required with Tender:**
(*a*) Outline drawing of plant erected, figured in with overall dimensions and pipe sizes.
(*b*) ....
(*c*) ....

11. **Technical Information** (to be filled in by Tenderer) :

(*a*) *Performance Details :*

Guaranteed to :—Remove......per cent. of all solid matter.

Remove all trace of free moisture and fog ?

Cool to......per cent. of the entering wet bulb depression.

Humidifying efficiency......per cent.

Gals. of water circulated per hour........

Pressure on spray nozzles......lbs./sq. in.

Pressure on scrubbing plate nozzles......lbs./sq. in.

B.H.P. of motor required.........

(For motor details see separate sheet.)

Friction loss between intake and discharge at rated duty......ins. W.G.

(*b*) *Gross Weight* of plant will be :

Washer complete .... lbs.
Pump and connecting pipes .... lbs.
Water in settling tank .... lbs.
Water in pump and pipes .... lbs.
Total weight as fixed for operating .... lbs.

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....
....
....
....
....

Signed..............
Dated.........

Trade Description No. 4.

PLENUM HEATER.

[*Strike out words in italics which do not apply.*]

Job No.

1. Inquiry to Messrs.......................................
....
....

2. Delivery required to site at...........carriage free on .../.../19.........certain/*and erected complete, ready for beneficial use by*.../.../19.........*certain*. (Usual for fan, heater, and boiler contracts to be placed together.)

3. **Heading Medium:**
Steam at 5 lbs. boiler pressure/Forced hot water at 190° *Fahr.*

4. **Capacity** at......ft./min. air flow through free area.
(*a*) Tempering coil ....B.Th.U.
(*b*) Re-heating coil ....B.Th.U.
Total . . ..B.Th.U.

5. **Type:**
To be of the *Sectional Cast Iron/Wrought Iron Pipe*/type, built up into stacks and connected to main headers, as shown on Drawings No.......and approved by the Engineer.

6. **Supports:**
To be of rolled steel generally, as shown on the Drawing No......., supplied and erected, by Contractor.

7. **Maximum External Dimensions.** (Free area through heater varies from 37 to 61 per cent. of gross area. Velocity of air through free area 600/1,500 ft./min.)
(*a*) Height of......ft.......ins. } Cross - sectional area of
(*b*) Width of......ft.......ins. } ......ft. super.

8. **Distance Apart:**
Length from discharge side of tempering coil to inlet side of re-heating coil to be......ft.......ins.

9. **Valve:**
To be provided on flow and return of each stack, etc.

10. **Drawings Required with Tender:**
(*a*) Outline drawing of complete heater figured in with overall dimensions and pipe sizes.
(*b*) ....
(*c*) ....

11. **Technical Information:**

	Tempering Coil.	Re-heating Coil.
(*a*) Consumption of heating medium will be per hour		
(*b*) Area of heating surface in feet super.		
(*c*) Number of stacks		
(*d*) Number of sections per stack		
(*e*) Frictional loss in air circuit in inches W.G.		
(*f*) Friction loss in heating circuit feet of water col./lbs. per square inch...		

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....
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....

Signed..............
Dated........

Trade Description Sheet No. 5.

DIRECT CURRENT MOTOR.

(Silent Running, Essential.)

[*Strike out the words in italics which do not apply.*]

Job No.

1. Inquiry to Messrs..
....
....

2. Delivery required to site at...........carriage paid by .../.../19.........certain/*and erected complete, ready for beneficial use by.../.../19.........* (Unusual for motor manufacturers to carry out erection.)

3. **Electricity Supply:**
Continuous at......volts.

4. **Output of Motor:**
(*a*) ......B.H.P. *Continuous/Crane/*rated.
(*b*) Overload capacity required.........(usually 25 per cent. for one hour, and 50 per cent. for one minute).
(*c*) Speed : *Constant/Variable/*......R.P.M. to......R.P.M.

5. **Type :**
*Open/Semi/Totally/*enclosed.

6. **Winding:**
*Shunt/Series/Compound/*wound.

7. **Rotation:**
To be *clockwise/counter clockwise/*when viewed from the commutator end.

8. **Purpose required for:**
*Belt/Chain/Direct/*coupled to a.........

9. **Working Position:**
   (*a*) Motor Shaft/*Horizontal*/*Vertical.*
   (*b*) Magnet feet on *floor*/*ceiling*/*wall.*

10. **Pulley:**
   Flat/*Convex*/of......inches diam. by......inches width, having crowning of......inches (Flat, double width required for fast and loose pulley drive).

11. **Switchgear:**
   (*a*) Starting switch to be of the *open*/*semi-enclosed*/*totally enclosed*/*ordinary*/*slow motion*/type, fitted with overload and no-voltage releases, by Messrs...............as Cat. No.......
   (*b*) Ammeter of moving *iron*/*coil*/type......inches diam. dial by Messrs...............as Cat. No.......

12. **Terminal Box:**
   To be suitable for V.I.R. cables and screwed ......inch electric thread conduit.

13. **Spares and Sundries :**
   (*a*) Complete set of spanners to fit every nut on the machine.
   (*b*) Box spanner for stud in pulley boss (pulley face to be drilled to pass same if necessary).
   (*c*) Tommy bar for forcing screws of slide rails, and for box spanner.
   (*d*) Angle iron rack for above.
   (*e*) Complete spare set of commutator brushes.
   (*f*) Complete spare set of bearing bushes.
   (*g*) Slide rails.

14. **Drawings Required with Tender:**
   (*a*) Outline drawing of complete machine figured in with overall dimensions and pulley centres.
   (*b*) Speed-load and efficiency curves.
   (*c*) ....

15. **Technical Information** (to be filled in by Tenderer) :
   (*a*) Efficiency will be :—
   ......per cent. at $\frac{1}{2}$ load ; ......per cent. at $\frac{3}{4}$ load ;
   ......per cent. at full load ; ......per cent. at $1\frac{1}{4}$ load.
   (*b*) Consumption at full load will be......B.T.U./hour.
   (*c*) The commutator will be......inches long and will contain......segments.
   (*d*) The brushes we propose to use will be......grade as made by the..............Co.
   (*e*) Gross weight will be......lbs.
   (*f*) The armature will be......inches diam. and will have ......slots, *open*/*semi*/*totally*/enclosed.

(*g*) Commutating poles *will/will not*/be fitted.
(*h*) Shunt field current will be......min.......max. amps.

....
....
....
....
....
....

Signed..............
Dated.........

Trade Description Sheet No. 6.

ALTERNATING CURRENT MOTOR.

(Silent Running, Essential.)

[*Strike out the words in italics which do not apply.*]

Job No.

1. Inquiry to Messrs...
....
....

2. Delivery required to site at...........carriage paid by .../.../19.........certain/*and erected complete, ready for beneficial use by.../.../19*......... (Unusual for motor manufacturers to carry out erection.)

3. **Electricity Supply:**
(*a*) *Single/Two/Three*/phase.
(*b*) Voltage......volts.
(*c*) Periodicity.........

4. **Output of Motor:**
(*a*) ......B.H.P. *Continuous/Crane*/rated.
(*b*) Overload capacity required......per cent., usually 25 per cent. for 30 mins. and 50 per cent. momentarily.
(*c*) ......speed.
(N.B.—Table XXII.)
(*d*) Power factor to be not less than 0.......at full load.

5. **Type:**

*Open/Semi-Enclosed/Totally/*enclosed.

6. **Rotor Winding:**

Slip ring/Short circuit. (Latter unusual above 2 B.H.P. on lighting supply circuits.)

7. **Starting:**

(*a*) Torque: *On light load* (*against full load*).
(*b*) Current: Limited to......times full load current.

8. **Purpose Required for:**

*Belt/Chain/Direct/*coupled to a.........

9. **Working Position:**

Motor shaft/*horizontal/vertical.* Magnet feet on *floor/ceiling/wall.*

10. **Pulley:**

*Flat/Convex/*of......inches diam. by......inches width, having crowning of......inches. (Flat, double width for fast and loose pulley drive.)

11. **Switchgear:**

(*a*) Starting switch to be of the *Open/Semi-enclosed/Totally enclosed/Ordinary/Slow motion/*type, fitted with overload and no-voltage releases, by Messrs................as Cat. No..........

(*b*) Ammeter of moving iron type......inches diam. dial by Messrs................as Cat. No..........

12. **Terminal Box:**

To be suitable for......V.I.R. cables and screwed......inch electric thread.

13. **Spares and Sundries:**

(*a*) Complete set of spanners to fit every nut on the machine.

(*b*) Box spanner for stud in pulley boss (pulley face to be drilled to pass same if necessary).

(*c*) Tommy bar for forcing screws of slide rails and for box spanner.

(*d*) Angle iron rack for above.

(*e*) Complete spare set of slip-ring brushes (wound rotor only).

(*f*) Complete set of bearing bushes.

(*g*) Slide rails.

14. **Drawings Required with Tender:**

(*a*) Outline drawing of complete machine figured in with overall dimensions.

(*b*) Speed-load curve.

15. **Technical Information:** (to be filled in by Tenderer) :

(*a*) Efficiency will be :—

......per cent. at $\frac{1}{2}$ load ; ......per cent. at $\frac{3}{4}$ load ; ......per cent. at full load ; ......per cent. at $1\frac{1}{4}$ load.

(*b*) Consumption at full load will be......B.T.U./hour.

(*c*) The brushes (slip-ring type rotor) we propose to use will be............as made by the..............

(*d*) Gross weight will be......lbs.

(*e*) The maximum rotor current (wound rotor only) will be......amp. per phase.

....

....

....

....

....

....

Signed..............

Dated.........

PART II.

DETAILED SPECIFICATIONS.

Detailed Specification No. 1.

SHEET METAL VENTILATING DUCTS.

To be erected at

....

1. Construction.

To be constructed and installed in accordance with Drawings No.......in positions approved by the Engineer.

2. Sizes and Delivery.

All special dimensions required to be taken from finished brick-work sizes on job. Deliveries to be made, and ducts fixed as and when required by the Engineer, as structural work proceeds.

3. Metal.

Make all sheet metal ducts of best quality galvanised steel sheets, with slip joints *in the direction of* the air flow. Rectangular ducts to have standing seams, and wide ducts to be stiffened by external angle irons.

4. Sizes and Gauges.

Gauge	28	26	24	22	20	18	16
Round, diam. ins.	4–8	10–18	21–24	27–33	36–45	50–60	65 & above
Rectangular, wide ins.	3–15	18	19–30	31–60	61–118	118 & above	—

5. Bends.

To be made in not less than five pieces, the radius of throat being equal to two diameters of pipe and in no case less than $1\frac{1}{2}$ ditto.

6. Increase and Decrease of Section.

Abrupt changes in section to be avoided in every instance. The length of the nozzle to be made as long as circumstances permit, preferably ten times the slope (= 12°), and in no case less than five ditto (= 6°).

7. Joints.

All joints to be smooth and tight, and all pipes to be firmly hung and rigidly fastened in place.

8. Branches.

When possible, to be tapped off from the main pipe at an angle of 30° and turned on a radius of throat equal to two diameters. In restricted positions the two diameter radius of throat may be used only.

Each branch rectangular duct to be provided with a damper and quadrant, which may be set and locked in position.

Round branch outlets to have adjustable butterfly dampers.

9. Adjustment.

After erection, test and set dampers for proper air distribution as directed by the Engineer. All duct joints to be tight at static pressure equivalent to 2 inches of water gauge.

10. General.

All work and materials to be of the best of their respective kinds, free from faults and flaws, and to the satisfaction of the Engineer.

No variation from the standard bends, branches, converging and diverging nozzles, etc., specified will be accepted and paid for unless written permission had been previously obtained from the Engineer for each specific instance.

11. Standard Designs.

All branches, increases, and decreases of section, bends, dampers, etc., to be in accordance with Standard Drawing No.......attached hereto.

12. Cleaning.

Ample provision to be made for cleaning all plenum ducts either directly or by means of flue brushes.

Detailed Specification No. 2.

MULTIBLADE CENTRIFUGAL CASED FAN.

To be supplied, delivered, and erected at

....

The specification following indicates the requirements and preference of the purchaser in the matter of detail.

At the same time it is recognised that excellent designs may differ materially from that herein outlined; therefore tenders will be considered, even although the specification is not strictly complied with, if the specified performances can be guaranteed, the deviations therefrom clearly set forth in the space provided, and the proposed plant be seen working satisfactorily in London.

[*Delete words in italics which do not apply.*]

1. Size and Capacity.

Supply, deliver, and erect where shown on Drawing No....... on purchasers' foundations one multiblade (*¾/full*) housing (*top/bottom horizontal/vertical up/down*) discharge (*single/double*) width, motor or pulley on (*right/left*) hand when looking into discharge, having a capacity of......cubic feet of air per minute when measured at a temperature of 60° Fahr., delivered against a maintained (static) pressure of......inches water.

2. Housing.

To be constructed, in......parts for convenient handling on site, of best commercial steel plate with riveted lap seams, braced by vertical and horizontal angle irons, and fitted with angle iron base frame drilled for holding down bolts. (Include for diverging nozzle on discharge having length equal to ten times slope and final area of......sq. feet.)

3. Suction Eyes.

For double inlet fan.

The wheel to be composed of two separate single width wheels mounted back to back, having common blade plate. Each/The inlets to be fitted with an inlet cone in the space between housing and wheel, having a minimum clearance with the flanged inlet of the runner.

4. Runner.

To be of the curved multiblade type, having blades of steel plate riveted at the back to a boiler plate disk, which in turn is to be hot riveted to a cast-iron hub. These blades to be connected by a flange at the inlet edge of the wheel.

Hub is to be attached to the shaft by key and set screws, and to the inlet flange by four heavy tie rods.

Runner to be carefully balanced to prevent vibration.

The inner edge of the blades to be so arranged as to give a decreasing inlet diameter from front to back in order to give a uniform radial velocity through the wheel. The angle of the blades to vary across the width in order to ensure the entrance of the air with the least possible loss by shock.

The curvature of the blades to be such that at normal or rated capacity the air will leave the tips with a velocity pressure approximately twice the pressure corresponding to the peripheral velocity of the wheel, in order to reduce the required speed of rotation.

5. Shaft.

To be of the highest grade forged mild steel of substantial size, machined all over.

6. Bearings.

To be spherical self aligning, ring lubricated type, of cast iron, lined with best quality babbit, length at least two and a half times diameter of shaft, and so designed as to allow easy adjustment for wear. Bearings to be provided with large oil reservoirs, fitted with suitable level indicators. In the case of a bearing mounted in the fan inlet, it is to be provided with suitable arrangements for

preventing oil from being drawn along shaft and into the fan by the entering air.

Inspection-hole lids to be attached by short lengths of substantial chain to the pedestal.

7. Thrust Collars.

To be smoothly turned and work against babbited shoulders on the ends of the bearings in order to avoid a thumping noise, due to uneven collar, bearing ends, or variations in the air current.

8. Pulley.

Include for......inches diam. cast iron fast (and loose) pulleys having an unequal number of curved arms. Width of face to be ......inches, and crowning of...... inches.

9. Label.

An engraved or cast-brass label to be securely fixed to the fan stating clearly :—

(*a*) Maker's name.
(*b*) Year of manufacture.
(*c*) Maker's serial number.
(*d*) Maker's catalogue size.
(*e*) Rated speed in R.P.M.
(*f*) Rated output in cubic feet per minute.
(*g*) Static pressure in inches.
(*h*) Rated B.H.P. for above conditions.

10. Painting.

Casing, runner, and pedestals to be painted two coats of good oil paint before delivery, and one after erection.

TENDERER'S DETAILS.

No Tender will be considered unless the full details are filled in.

1. Maker.........
2. Duty......cubic feet per minute.
3. Static pressure......inches water.
4. Area of suction......sq. feet.
5. Area of discharge......sq. feet.
6. Velocity through discharge......feet per minute.
7. Corresponding velocity head......inches water.
8. Total pressure (static plus velocity)......inches water.
9. Ratio of static pressure to rated total pressure......per cent.
10. Gauge of plate for casing......B.W.G.

11. Total weight of casing, runner, shaft, bearings, pedestals, pulley, etc., etc.......lbs.
12. Guaranteed mechanical efficiency of fan :—
    (*a*) *Static* :—

$$\frac{\text{A.P.M.} \times 0{\cdot}000157 \times \text{static head in ins.}}{(\text{B.H.P. at motor pulley} - 3 \text{ p.c. for belt loss})} = \ldots\ldots\text{p.c.}$$

(N.B.—Static pressure is defined as the arithmetical sum of the static readings on the two sides of the fan minus the velocity pressure at the fan inlet.)

(*b*) *Total or Dynamic Efficiency* :—

$$\frac{\text{A.P.M.} \times \text{total pressure in ins.} \times 0{\cdot}000157}{(\text{B.H.P. at motor pulley} - 3 \text{ p.c. for belt loss})} = \ldots\ldots\text{p.c.}$$

(N.B.—The velocity pressure is defined as the pressure corresponding to the average velocity over the area of the inlet or outlet of the fan, whichever is the higher. Total pressure = velocity plus static pressure.)

Detailed Specification No. 3.

SPECIFICATION

for the Supply, Delivery, and Erection of

AN AIR WASHER AND HUMIDIFIER

The specification following, indicates the requirements and preference of the purchaser in the matter of detail.

At the same time it is recognised that excellent designs may differ materially from that herein outlined, therefore Tenders will be considered even although the Specification is not strictly complied with if the specified performances can be guaranteed, the deviations therefrom clearly set forth in the space provided, and the proposed plant be seen working satisfactorily in London.

1. Extent of Contract.

Include for the supply, delivery, erection on purchasers' foundations, setting to work, and testing of one Air Washer and Humidifier complete, consisting of casing, tank, eliminator, spray system, motor-driven circulating pump, strainers, inter-connecting pipework, gauges, and valves in position as shown on Drawing No.......

2. Duty and Requirements.

To be suitably designed for handling continuously......thousand cubic feet of air per minute at a maximum velocity through the washer of 500 feet per minute, and a guaranteed maximum resistance of 0·375 inch water.

When operating under these conditions the washer shall—

(*a*) Entirely remove all free moisture and entrained water from the treated air.

(*b*) Remove 98 per cent. of the solid material carried by the entering air.

(*c*) With the mist sprays and the flooding sprays operating and the eliminator in use the minimum humidifying efficiency to be 85 per cent.

(*Note.*—By humidifying efficiency is meant the ratio of the weight of moisture which the air actually absorbs in its passage through the washer to the weight which it would absorb to saturation.)

(*d*) Remove effectually all trace of fog from the treated air.

3. Casing and Tank.

Casing to be constructed of galvanised iron of 18 B.W.G., the leading dimensions being approximately—

......ft.......ins. high.
......ft.......ins. wide.
......ft.......ins. long overall.

Settling tank, at least 16 inches high, to extend under the entire washer and to be made of 16 B.W.G. galvanised iron.

Casing and tank to be braced on the outside with 1½-inch by 1½-inch galvanised angles, which shall not be spaced further apart than 3 feet.

The sides, top, and tank to be made separately and bolted up with rubber gaskets on site.

All rivet holes to be soldered over on inside of casing.

A perforated galvanised distributing plate at the inlet of the washer, having 50 per cent. free area, to be provided.

All metal work to be galvanised after cutting or punching.

The settling tank to be divided into two compartments by a 20-mesh copper gauze strainer rated at 1 sq. foot for each 4,000 cubic feet of air handled per minute, and through which the water passes before entering the suction of the pump. The top of the suction strainer compartment to be fitted with a hinged lid over the entire length, provision being made for easily cleaning and replacing of the copper gauze when necessary.

Pump suction pipe to be continued down to within 2 inches above bottom of tank to avoid air troubles.

4. Inspection Door.

Where shown on drawings, provide, in the sides of the washer casing, air and water tight doors of minimum dimensions, 15 inches by 24 inches, having lower edge 36 inches from floor level, to allow easy cleaning and convenient access to the machine.

Door to be of cast iron with two glass panels, each glass being not less than 9 inches by 12 inches. The door frame also to carry a $\frac{1}{4}$ inch by $\frac{1}{4}$ inch pure rubber gasket against which the door is to close.

Frame to be of cast iron and riveted through the washer casing to a companion frame on the inside. To be provided with gutter at lower edge for catching any leakage water and returning to the tank.

Doors to be held by at least three cams on each side, easily operated and fixed solidly against the gasket, and sufficiently rigid to prevent cracking of glass. Hinges to be so constructed as not to interfere with the action of the cams.

5. Tank Fittings.

(*a*) One $\frac{3}{4}$-inch automatic float valve, suitable for 160 lbs. per sq. inch supply, for maintaining constant water level.

(*b*) An independent $\frac{3}{4}$-inch full-way gate valve, with hose connection for quickly filling tank after cleaning.

(*c*) Sufficient stout rubber canvas hose, fitted with screw connector, to reach to the opposite side of the tank.

(*d*) A 2-inch flanged connection at the bottom of the tank, with elbow and brass gate valve for draining off the water.

(*e*) An independent bell mouthed 2-inch cast-iron overflow, connected outside the tank to a tee from which the connection to sewer is made under separate contract.

6. Spray System.

The system to consist of a series of brass spray nozzles evenly spaced over the entire area, and placed at least 4 feet back of the eliminator nozzles but in a parallel plane, thus uniformly filling the intervening space with a finely divided spray.

A minimum of 2·5 spray nozzles per 1,000 cubic feet of air handled per minute, producing a mist-like spray, to be provided.

Nozzles to have a discharge orifice not less than $\frac{3}{16}$ inch diameter, operating satisfactorily on a maximum pressure of 25 lbs. per sq. inch.

Stand pipes to be $1\frac{1}{4}$-inch galvanised extra heavy wrought-iron pipe screwed into a galvanised cast-iron header.

The flooding nozzles over the eliminators to be spaced on 3-inch centres and handle one gallon per minute each.

7. Scrubbing and Eliminating Plates.

The eliminators to be set in vertical position and be made of 24 B.W.G. galvanised iron. The angles of the eliminator to be not greater than 35° and so set that the air in passing through is deflected at least six times.

Eliminator plates to be bolted or riveted directly to galvanised iron supports.

The space for passage of air between any two adjacent eliminator plates not to exceed 1 inch.

No separate metal clips to be used.

The machine to be so arranged that the first four bends of the eliminator plates shall become a washing surface.

A separate set of sprays independent of the main sprays to be provided for maintaining a constant sheet of water flowing down these four surfaces continually.

Minimum washing surface to be 40 sq. feet per 1,000 cubic feet of air per minute.

The last two bends of the eliminator plates are to remove effectively all free and entrained moisture.

The total washing and eliminating surface shall not be less than 60 sq. feet per 1,000 cubic feet of air per minute.

8. Pump and Motor.

To be of the brass-fitted, double suction, centrifugal type, having a capacity of......gallons per minute when discharging against sufficient head to obtain perfect spray effect of all nozzles.

The pump to be of the horizontal type having enclosed runner, and to be provided with cast-iron base plate, with oil ring round same, for direct coupling to a......B.H.P. motor.

The casing of the pump to be of grey cast-iron, horizontally divided for convenient inspection, suitable to withstand an excess over the working pressure and designed with ample water ways for proper velocity.

The usual piping, drains, priming cock, foot valve, and grease cups, in addition to throttle valve, on discharge side, to be included for.

All surfaces not machined to be rubbed down, filled in smooth, and given two coats of good oil colour before delivery.

The pump and motor to be fixed on the purchasers' foundations, the bed-plate being wedged up, and when machines are in perfect

alignment to be properly grouted in and bolted down. Pipes to be fitted to pump, and not *vice versâ*.

Engraved or cast-brass label to be securely fixed to pump, stating concisely :—

(*a*) Maker's name.
(*b*) Year of manufacture.
(*c*) Maker's serial number.
(*d*) Speed in R.P.M.
(*e*) Rated output in G.P.M.
(*f*) Rated head in feet.
(*g*) Rated B.H.P.

Motor to be to standard specification attached herewith, to be direct coupled to pump by means of an approved flexible coupling. Dowel pins to be fitted to both pump and motor for rapid lining-up on re-erection.

9. Pot Strainer.

One pot strainer of suitable capacity and fitted with removable gauze basket to be provided, fixed in the discharge line running from the centrifugal pump to the spray header, and clear of the motor.

The baskets of this strainer to be made of 14-mesh copper wire cloth screen, giving an area of at least sixteen times the area of the pipe connection.

The top of this strainer to be held by clamps and screw made on rubber gasket joint between machined surfaces so that it can be instantly and easily removed and choked basket replaced with a clean one.

One spare basket to be provided.

10. Pipework, Valves and Gauges.

Include for all inter-connecting pipework in galvanised steel, with separate full-way brass gate valves, all of suitable capacity and by approved maker, to main sprays, flooding nozzles, water supply, and drain respectively. Water supply and drain connections laid on separate contract.

Also for two 3-inch best quality pressure gauges, scale reading 0–50 lbs., fitted with brass plug cocks and connections ; to be fixed in approved positions on the main spray system and on the flooding nozzles respectively.

All pipe connections to the inside of the washer to be provided with companion flanges and pure rubber gaskets on *each* side of the casing.

11. Take-over Tests.

(N.B.—It is recognised that these tests are by no means perfect, but are here put forward to and will form the basis of the contract in the absence of anything more satisfactory.)

(a) Free Moisture.

The washer is to pass no free moisture that will show on a plate of clean glass 6 inches square held for a period of ten minutes 8 inches back of the eliminator and at right angles to the direction of flow of the air.

(b) Cooling and Humidification (Maximum).

The apparatus, when recirculating water in summer and handling rated capacity of air, is to reduce the temperature of the air 70 per cent. of the entering wet bulb depression.

The apparatus, when using cold water in summer, to reduce the outgoing difference of air (wet bulb) and water temperatures to less than 25 per cent. of the difference in the incoming air (wet bulb) and water temperature.

(c) Humidification (Minimum).

When the washer is operating with the eliminator flooding nozzles only, the washer to reduce the temperature of the air 10 per cent. of the entering wet bulb depression as a maximum.

(d) Test for Coarse Dirt.

One lb. of dry boiler soot at a temperature not less than 70° Fahr. for every 20,000 cubic feet of air per minute to be sifted into the mouth of the washer at a uniform rate in one minute, and uniformly across the width of the washer.

The soot to be entirely washed from the air in passing through, as determined by a sheet of white paper not less than 6 inches square suspended at right angles to the flow of the leaving air. Test paper to be coated with a solution of white sugar syrup.

(e) Test for Fine Dirt.

Two cross-wire screens covered by a thin film of absorbent cotton not less than 6 inches square are to be placed as follows:—

One screen to be inserted in the air duct at any convenient point so as to intercept the air before it passes through the washer. The other screen to be placed in the air duct leading from the washer at approximately the same location in the cross-section of the duct and in the current of the air after it has been washed.

The first screen to remain in the air current for six hours continuously, and the second screen is to remain in the air current

continuously for two weeks, the entering air being kept in as uniform condition as possible as regards dust during the test. At the expiration of the two weeks the second screen is to show no greater collection of dust than the first screen. It is understood that the second screen is to be placed back of the washer a distance not more than the width of the duct, so that it will be unaffected by any air leaks in the duct between the air washer and the screen.

(*f*) *Test for Fog.*

Fog to be tested for in the ducts by means of an electric lamp projecting a parallel beam of light.

(*g*) *Noxious Odours.*

A burning smoke rocket, as used by the city health departments for testing drains, to be moved uniformly and slowly over the intake grating.

All tests, except when otherwise directed, to be made under the specified normal full load conditions.

12. Spares.

Two gauze buckets for strainer.

TENDERER'S DETAILS.

No Tender will be considered unless the full details are filled in.

1. Spray nozzles per 1,000 cubic feet of air handled.........
2. Water pressure on spray system for maximum efficiency ......lbs. per sq. inch.
3. Angle at which eliminating plates are set to normal direction of air flow......degrees.
4. Number of times air is deflected in passage through eliminator ....
5. Space between individual eliminator plates......inches.
6. Area of washing surface (as distinct from eliminator surface) per 1,000 cubic feet of air handled......sq. feet.
7. Maker of pump.........
8. Catalogue number.........
9. Speed of pump......R.P.M.
10. Maker of motor.........
11. Guaranteed mechanical efficiency at full load......per cent.
12. Electricity consumption in Board of Trade units per hour at full load.........
13. Maker of flexible coupling (if supplied).........

14. *Drawings required* :—

(*a*) Blue, with leading dimensions filled in, showing general arrangements.

(*b*) Blue, of pump and motor.

(*c*) Characteristic curves :

(A) Pump.

(B) Motor.

(C) Washer operating at various spray pressures.

15. *Variations.*

The apparatus put forward will completely comply with the specification, except as regards the under-mentioned details : –

....

....

Detailed Specification No. 4.

DIRECT CURRENT MOTOR.

Tenders will be considered even although the Specification is not strictly complied with, if similar results are obtained, *and any variations stated fully*, but SILENT RUNNING IS ESSENTIAL.

1. Output.

To be capable of continuously (six hours rating) developing the specified B.H.P. at the pulley, when running at any speed within the specified range.

Variable speed machines to be specially designed for stable running on weakened field at the higher speed.

2. Frame and Field Poles.

To be of the best close-grained cast iron, or cast magnet steel, into which the laminations are to be secured.

Eye-bolts to be securely fixed into magnet case, of under-mentioned minimum sizes :—

Up to 6 inches diam. armature, one for 1-inch tommy bar.

6 inches to 8 inches diam. armature, one for $1\frac{1}{4}$-inch tommy bar.

Above 8 inches diam. armature, two for $1\frac{1}{4}$-inch tommy bar.

The four poles, which are to be accurately spaced, are, together with their pole shoes, to be made up of charcoal iron laminations, riveted *tightly* together and securely fixed at *diagonal* points of the magnet case, or alternatively, the magnet case may be of cast iron or steel having cast poles and charcoal iron laminated tips, securely riveted together and screwed to same by at least two counter-sunk screws.

3. Field Coils (Shunt).

To be former wound, thoroughly varnished and baked. Afterwards, insulated with presspahn, well bound with damp-proof tape, well soaked in insulating compound and again baked for twenty-four hours. Each coil to be provided with its own terminals and securely fixed in position on the magnet poles.

4. Field Coils (Series).

Where fitted, to be of strip copper, and so formed that the starting and finishing ends of the coil are on the outside. Connections between different coils to be made of flexible cable or solid copper strip, insulated with prepared tape.

5. Armature.

Of the slotted drum type, built up of charcoal iron laminations, carefully insulated from each other and adequately ventilated by radial ducts.

Coils to be former wound, and fixed in the slots—previously lined with a trough made of approved insulating material and moulded into shape before being placed therein.

Insulation of coils to be protected from mechanical injury by turning over the insulating lining of slot to meet at the centre and covering by a further strip of tough damp-proof insulating material.

Armatures to be well bound with best quality tinned steel wire well soldered up, insulated from the core plates by mica and carried in recesses in the core so that in case armature rubs on poles shoes, the steel bands are not damaged.

6. Air Gap.

Clearance between pole tips and any part of armature to be not less than under-mentioned figures :—

Up to 8 inches diam. armature, 0·05 inch.

Above 8 inches and up to 12 inches diam. armature, 0·065 inch.

Above 12 inches and up to 18 inches diam. armature, 0·075 inch.

Above 18 inches, 0·065 inch.

7. Commutator.

To be carried on rigid sleeves positively driven from the shaft and built up of an approved number of hard drawn or drop forged copper segments, insulated throughout with pure Indian mica of *same wearing hardness as the copper.*

Segments to be of ample area, and of such radial depth that not

less than $\frac{1}{2}$ inch can be turned off radially without causing undue heating when the machine is giving its normal full load current continuously with the reduced section.

Mica to be chased out just clear of the commutator surface *before official test.*

8. Laminations.

Plates of the armature, and also the sleeve carrying the commutator, to be positively driven, special care being taken to prevent them working loose on the shaft from the hammering action due to intermittent loads, or from any other causes.

9. Balance.

Armature and commutator to be properly balanced and set in the magnet fields in such a way as to avoid end thrust and run truly whether field coils are excited or not.

10. Brush Gear.

Brush holders to be of box type, with readily adjustable tension spring giving wide range of even pressure on brushes. Brushes to be of some standard size made by the Morgan Crucible Co., Ltd., or other well-known maker, self-adjusting, box type (sample to be submitted and approved), and fitted with copper flexibles; both commutator and brush surface to be ample.

Machine to run sparklessly and without trace of "hunting" on weakest field (when driving under specified conditions), at all loads and speeds, with fixed brush position up to 100 per cent. in excess of the specified maximum output, and a suitable permanent index to be provided, showing clearly the position of the brush rocker for sparkless running.

Collector rings to be of solid copper strip, insulated with prepared tape and fixed at least $\frac{1}{2}$ inch clear of all parts of frame.

Brush boxes adjustable on rocker arm in order to give even wear on commutator.

11. Shaft.

To be of best wrought mild steel and of ample proportions to render it strong and stiff, accurately turned and ground true, and suitably designed to prevent oil creeping on to armature or commutator.

12. Bearings.

Bearings to be of ring lubricated, self-oiling type, of solid phosphor bronze. Ratio of length to diameter of shaft to be not less than

2·5 : 1. Grooves to be machined in bushes at either end to carry away excess oil, *and channels to be cut into inner surface of bush to ensure access of oil to all parts of shaft.* Approved oil depth gauges and run-off taps to be provided for each bearing. Efficient oil flirts to be provided on the shaft which absolutely prevent access of oil to the windings.

13. Pulley.

To be of cast iron of the specified dimensions. Centre hole to be a good sliding fit on shaft.

Pulley to be secured to shaft by substantial feathers held in place by suitable stud, screwed through boss.

14. Slide Rails.

Motor to be bolted down to two heavy cast-iron single-type slide rails amply proportioned to render them strong and stiff, accurately planed on top surface, and each provided with bolt hole lugs for securing to concrete bed.

Forcing screws to have coned ends (where they engage magnet case), cylindrical heads with tommy bar holes, and allow minimum travel of under-mentioned figures when fixed at opposite ends of magnet case.

Size of Motor.	Travel of Magnet Case (Inches).	Rag Bolt Holes (Inches).
Up to and including 5 B.H.P. .	3	1
Up to and including 10 B.H.P.	6	$1\frac{1}{8}$
Above 10 B.H.P. . . .	9	$1\frac{1}{4}$

15. Terminals.

Each circuit, viz., shunt field, series coils and brush gear to be brought out on to separate terminals fixed on approved enamelled slate—minimum thickness $\frac{3}{4}$ inch—which is to be efficiently protected by a heavy cast-iron box lined with insulating material, into which a metallic flexible tube can be screwed direct.

No cables to be visible when the cover plates are in position, terminal box being designed so that all connections are brought into the back of the terminal box and not at the sides or bottom, with loops of unprotected cable.

Sweating thimbles of approved design to be provided for external

connections and copper strips of suitable proportions to be provided for internal connections. Lock nuts or two set screws to be fitted to each terminal. Earthing stud and thimble to be fixed to slide rails.

16. Maker's Name Plate.

An engraved brass plate to be securely attached to the side of magnet case, the following particulars to be clearly stamped thereon :—

(*a*) Maximum B.H.P. (continuous rating).
(*b*) Voltage of circuit for which motor is wound.
(*c*) Full load current in amperes.
(*d*) Speed in R.P.M. minimum and maximum.
(*e*) Maker's name, reference number and year.
(*f*) Shunt field current (minimum and maximum) in amperes.

17. Temperature Rise.

The construction of the whole machine to be such that after running continuously upon its maximum output for at least six hours the temperature of the machine, or any part of it, shall not exceed 70° Fahr. above the temperature of the surrounding atmosphere, provided that in no case shall the total temperature exceed 140° Fahr.

18. Silent Running.

All parts of the machines must be thoroughly secured, *special precautions being taken to prevent any hum or other objectionable noise being given out at any load.*

19. Starting Switch.

To be of the specified type fitted with overload and no voltage release coils, and to be designed to start the motor up on *full load*, without undue heating. Temperature rise of such coils in no case to exceed 70° Fahr. above the temperature of the surrounding air after six hours' run at full load.

Slate panels carrying contracts to be rendered thoroughly non-hygroscopic. Where fixed in frame, holes in slab to be well bushed and provided with mica washers back and front.

Resistance coils to be either :—

(*a*) Wound on tubes formed of non-hygroscopic and non-brittle material so fixed as to have ample room for expansion due to heating.

(*b*) Spiral coils, supported on slate slabs and fixed well clear of the case and of adjacent coils.

No solder to be used and all terminals to be fitted with lock nuts.

Rheostat arm to be fitted with adjustable tension plate and renewable contact brushes. Field and first armature contacts to be carbon, and fixing screws (if any) on face of slab to be protected by ebonite caps or hard wood plugs.

Separate earthing stud and thimble to be fixed on outer case and sweating thimbles provided for external circuits.

20. Field Regulator.

The separate rheostat to be provided for speed regulation on the shunt field, to conform generally to par. 19, to be designed to give the variation of speeds as specified, to be fitted with the specified number of contact studs, and to be so arranged that motor cannot be started up on weakened field.

Cover to be marked with suitable permanent indexes "SLOW" and "FAST," the movement of radial arm when speeding up being CLOCKWISE. Indicator to be provided outside the case to show exact position of contact arm.

21. Insulation Resistance and Pressure Test.

After six hours' test at full load, all windings, switchgear, etc., to be connected together and insulation resistance, copper to earth, when tested at 500 volts, to be not less than two megohms.

The under-mentioned pressure is then to be applied for two minutes between copper and stampings and between copper and binding wire and the 500 volt insulation test, as above specified, again repeated.

For machines up to 250 volts, test pressure 1,000 volts A.C.

For machines above 250 volts and up to 600 volts, test pressure 2,000 volts A.C.

22. Official Inspection and Take-over Test.

Contractor to give the Purchaser seven days' notice in writing of the date fixed by him for the official inspection and take-over tests of this machine, *together with all switchgear and accessories.*

The Purchaser's Inspector to be given necessary facilities, current, loan of instruments and assistance, to test the machine in order to prove that each part complies with the requirements of this specification.

In the event of rejection of the machine or switchgear by the Inspector, due to its failure to meet the requirements of this specification, a second inspection and test by the Inspector will be necessary. In this event the delay in delivery will continue to be the subject of the penalty clause contained in the Form of Tender, and the

Contractor will be charged the third-class rail and reasonable out-of-pocket expenses incurred by the Inspector in consequence.

23. Painting.

Outer surface of castings, etc., to be carefully filed flat, filled into a dead smooth surface, rubbed down, and painted two coats of dark green oil paint.

24. Guarantee.

Contractor to undertake for a period of twelve months from date of starting to run, to make good any original defects of design, material or workmanship, which may become apparent in use, free of all cost, excepting such as are due to ordinary fair wear and tear, fire, frost, wilful or accidental damage or other causes beyond his control, and on the assumption that all the machinery is properly looked after, kept clean, and lubricated.

It being understood that any faulty part will be returned to Contractor's works, carriage paid, and Contractor to have the option of replacing or repairing the defective parts.

Consequential damages excluded.

25. Payments.

Accounts, less $2\frac{1}{2}$ per cent. cash discount, to be paid on the 20th of the month *following date of delivery of the machine, together with all switch gear, etc., on site*, provided the order number is quoted on all invoices and statements and the accounts are rendered correctly and promptly.

26. Empties.

The Purchasers will not be responsible for the return of or payment for empties, but will put on rail carriage paid, and will notify the Contractor details of serial numbers, nominal value, route of despatch, carter's name signing for same, etc., of empties that are *clearly stamped with their value and with the name of the Contractor*, etc., and assist so far as possible to secure their return to the Contractor.

CHAPTER XV

THE APPLICATIONS OF WET AIR FILTERS TO THE COOLING OF ELECTRICAL MACHINERY

1. Mechanical and Electrical Losses Converted to Heat.

In Chapter I. it was shown that the losses in the conversion of energy from one form into another almost invariably reappear in the form of heat at the rate of 3,400 B.Th.U. per kelvin or Board of Trade unit, or 2,530 B.Th.U. per horse-power lost, and it is this question of heating which in recent years has practically determined the limiting output of any particular size of electrical machine.

Designers have recognised this limit and have succeeded to a small extent in pushing it a little further away, *i.e.*, increasing the output per pound weight of metal in the machine, either by fitting fans to the rotating parts or by increasing the radiating surface of the stationary parts by adding fins or gills. This practice answered well enough when machines were of limited size, generating relative small quantities of heat, and requiring relatively small quantities of air, but when the output capacity began to increase, until now a 30,000 kilowatt set is becoming more or less common, external means for supplying the air have had to be utilised.

Example 20.—Consider a 5,000 kilowatt machine running at an efficiency of 95 per cent. (electrical output/shaft input). Thus the electrical loss is (5,000 × 5) 250 kilowatts, producing (250 × 3,400) 850,000 B.Th.U. per hour.

2. Quantity of Air Required.

The usual temperature rise allowed of 70° Fahr. on the small machines is reduced to approximately 35° to 40° Fahr. on the larger sizes. Therefore the air required for efficient

cooling of the 5,000 kilowatt set under consideration would be

$$\frac{850{,}000 \times 53}{40 \times 60} = 18{,}700,$$

say 19,000 cubic feet of air per minute, or 1,140,000 cubic feet of air per hour.

The practice of different makers varies slightly, but from information kindly supplied by the British Thomson-Houston Co., Ltd., the British Westinghouse Electric and Manufacturing Co., Ltd., and the General Electric Co., Ltd., it would appear that standard allowance is approximately 5 to 6 cubic feet of air per kilowatt *capacity* of plant, or 100 to 120 cubic feet of air per kilowatt lost per minute, corresponding to a temperature rise of approximately 35° to 40° Fahr. or 18° Cent.

3. Static and Velocity Pressures Required.

The frictional resistance to the passage of air through the machine varies largely, of course, with the type and size and the method of passing the air through the machine, but the limits may be stated safely at 3 inches minimum, up to 7 inches W.G. maximum, in some cases being as low as $2\frac{1}{2}$ inches W.G. with axial ventilation on a 3,000 kilowatt set.

As regards the velocity at which the air is put through the machine, this is of the order of 1,500 to 2,000 feet per minute, so that the connecting ducts can conveniently be figured on this basis.

The warm air may be discharged either into the engine-room or, from considerations of fire risk, connected to a duct communicating directly with a safe place outside, or if it is desired to utilise the heat contained, arrangements may be made for passing it into the boiler house, if this is conveniently near.

Referring to the fire risk, it is usual to provide a substantial damper which can be closed from a convenient place, so that in the event of a serious fault developing in the machine the air supply can be entirely cut off.

4. Wet Air Filtering.

Obviously, special precautions must be observed in dealing with these large volumes of air to avoid the presence of grit or oil, which would permanently injure the insulation and dust, which will rapidly choke the comparatively small air passages. As indicated in Chapter VI., wet-air filters are admirably adapted for this work.

The only possible objection that can be raised is the introduction of "moist" air into the windings of the machine, but this falls to the ground at once when it is remembered that the temperature of the air is *raised* immediately it enters the machine windings, and *the relative humidity falls* in a corresponding ratio. With any well-designed filter every trace of free moisture is entirely removed on the eliminating plates, but if any doubt is still entertained on this score the duct between the end of the filter and the inlet to the machine can be made a minimum length of 25 to 30 feet long, thus giving time for any minute traces of water carried over to evaporate.

It is said that slightly higher efficiencies are obtained when the power for forcing the air through the machine is supplied externally than when the fan blades form an integral part of the rotating part, but the writer has no figures to prove this.

The cooling effect obtainable when operating wet-air filters on dry days (*i.e.*, low relative humidity) is of course an additional advantage, increasing to a small extent the output capacity.

5. Temperature Recording.

The application of the electrical thermometer (Chapter IX., par. 1) would appear to have a useful scope in connection with large electrical units in which the "bulb" can be built into the machine and the temperature periodically obtained by the switchboard attendant.

TABLE XXXVI. — **Pressure and Horse-power Lost by Friction of Air in Galvanised Iron Pipes 100 Feet Long** (by the courtesy of the Sturtevant Engineering Co., Ltd.).

Diam. of Pipe in Inches.	Loss of Pressure and Horse-power.	Velocity of Air in Feet per Minute.							
		1,000.	1,200.	1,400.	1,600.	1,800.	2,000.	2,400.	2,800.
12	Ins. of Water	0·159	0·229	0·312	0·408	0·517	0·638	0·920	1·25
	H.P. Friction	0·019	0·034	0·054	0·081	0·11	0·15	0·27	0·43
18	Ins. of Water	0·107	0·154	0·208	0·273	0·345	0·426	0·613	0·719
	H.P. Friction	0·029	0·051	0·081	0·121	0·173	0·238	0·411	0·522
24	Ins. of Water	0·079	0·116	0·157	0·205	0·258	0·319	0·460	0·626
	H.P. Friction	0·039	0·068	0·108	0·162	0·231	0·317	0·548	0·870
30	Ins. of Water	0·064	0·091	0·126	0·164	0·207	0·255	0·367	0·500
	H.P. Friction	0·0496	0·085	0·136	0·203	0·289	0·396	0·527	0·685
36	Ins. of Water	0·053	0·076	0·103	0·136	0·173	0·212	0·307	0·417
	H.P. Friction	0·595	0·102	0·163	0·243	0·346	0·475	0·822	1·30
40	Ins. of Water	0·048	0·066	0·093	0·122	0·155	0·191	0·276	0·376
	H.P. Friction	0·066	0·114	0·181	0·270	0·385	0·528	0·913	1·45
44	Ins. of Water	0·043	0·062	0·085	0·119	0·141	0·174	0·253	0·342
	H.P. Friction	0·072	0·125	0·199	0·293	0·424	0·581	1·00	1·59
48	Ins. of Water	0·039	0·057	0·078	0·102	0·129	0·160	0·231	0·312
	H.P. Friction	0·079	0·137	0·217	0·324	0·462	0·634	1·09	1·74
52	Ins. of Water	0·036	0·053	0·072	0·095	0·119	0·146	0·212	0·289
	H.P. Friction	0·085	0·148	0·236	0·352	0·501	0·687	1·18	1·88
56	Ins. of Water	0·034	0·050	0·067	0·088	0·110	0·136	0·205	0·269
	H.P. Friction	0·092	0·159	0·253	0·379	0·540	0·74	1·27	2·03
60	Ins. of Water	0·033	0·047	0·062	0·081	0·104	0·128	0·185	0·250
	H.P. Friction	0·099	0·171	0·272	0·406	0·578	0·793	1·37	2·17

TABLE XXXVII.—**Air Velocities and Equivalent Pressures (in Inches) of Water Calculated at 60° Fahr.**

Velocity in Feet per Minute.	Inches of Water.	Velocity in Feet per Minute.	Inches of Water.
100—500	0·01	2,700	0·45
600	0·02	2,800	0·50
700—800	0·03	2,900	0·53
900	0·04	3,000	0·57
1,000	0·06	3,100	0·60
1,100	0·08	3,200	0·64
1,200	0·10	3,300	0·68
1,300	0·11	3,400	0·71
1,400	0·11	3,500	0·78
1,500	0·13	3,600	0·80
1,600	0·15	3,700	0·86
1,800	0·19	3,800	0·90
1,900	0·20	3,900	0·95
2,000	0·23	4,000	1·00
2,100	0·25	4,500	1·28
2,200	0·29	5,000	1·55
2,300	0·31	5,500	1·85
2,400	0·35	6,000	2·20
2,500	0·40	7,000	3·00
2,600	0·41	—	—

Slight discrepancies between various published tables are due to the temperature at which the specific gravity of air is calculated.

Table XXXVIII.—**Properties of Saturated Steam.**

Total Pressure in lbs. per sq. inch (measured from a Vacuum).	Gauge Pressure from Atmosphere (assumed at 15 lbs. per sq. inch).	Temp. in Degrees Fahr. of Steam.	Latent Heat in B.Th. U.
1	−14 Vacuum.	101	1,034
2	−13	126	1,021
3	−12	141	1,012
4	−11	153	1,005
5	−10	162	1,000
6	− 9	170	995
7	− 8	176	991
8	− 7	182	987
9	− 6	188	984
10	− 5	193	981
12	− 3	201	976
14	− 1	209	971
15 (14·7)	(atmosphere)	212	967
16	1	216	965
18	3	222	963
20	5	228	959
22	7	233	956
24	9	237	952
26	11	242	949
28	13	246	947
30	15	250	944
32	17	254	941
34	19	257	939
36	21	260	937

TABLE XXXIX.—**Weight of Round Galvanised Steel Pipe and Elbows of the Proper Gauge (U.S.A.) for Heating and Ventilating Systems.**

American Gauge.	Wt. per sq. ft. in lbs.	Equivalent S.W.G. (English).	Diam. of Pipe.	Area in sq. ins.	Area in sq. ft.	Wt. per Running Foot.	Wt. of full Elbow.
28	0·78	29·5	4	12·6	—	1·1	0·9
			6	28·3	0·196	1·4	1·7
			8	50·3	0·348	1·9	2·9
26	0·91	27	10	78·5	0·546	2·7	5·3
			12	113	0·782	3·2	7·6
25	1·03	26	15	176	1·22	4·5	13·5
			18	254	1·76	5·3	19·1
24	1·16	25	21	364	2·53	7·0	29·6
			24	452	3·14	8·0	38·6
22	1·41	22·5	27	572	3·97	10·9	59·1
			30	706	4·91	12·2	73·4
			33	855	5·92	13·5	88·9
20	1·66	21	36	1,017	7·00	17·2	124
			42	1,385	9·58	20·1	168
18	2·16	19	48	1,809	12·5	29·8	286
			54	2,290	15·9	33·6	363
			60	2,827	19·6	37·4	448
16	2·66	17·5	66	3,421	23·8	50·5	666
			72	4,071	28·2	55	793

(See also Chapter IV., par. 1.)

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TABLE XL.—Hygrometric Tables. Relative Humidity, Dew Points, and Grains of Moisture per Cubic Foot.

(Barometer Pressure, 30 Inches. Temperatures in Degrees Fahr.).

Dry Bulb Temperature.	Grains per Cubic Foot at Saturation.	Degrees Wet Bulb Depression, ° Fahr.																																												
		1			2			3			4			5			6			7			8			9			10			11			12			13			14			15		
		Per cent. Relative Humidity.	Dew Point Temperature.	Grains per Cub. Ft.	% R. H.	D. P. T.	G. C. F.	% R. H.	D. P. T.	G. C. F.	% R. H.	D. P. T.	G. C. F.	% R. H.	D. P. T.	G. C. F.	% R. H.	D. P. T.	G. C. F.	% R. H.	D. P. T.	G. C. F.	% R. H.	D. P. T.	G. C. F.	% R. H.	D. P. T.	G. C. F.	% R. H.	D. P. T.	G. C. F.	% R. H.	D. P. T.	G. C. F.	% R. H.	D. P. T.	G. C. F.	% R. H.	D. P. T.	G. C. F.	% R. H.	D. P. T.	G. C. F.	% R. H.	D. P. T.	G. C. F.
30	1.94	89	27	1.72	78	25	1.50	67	21	1.30	56	18	1.08	46	14	0.89	36	8	0.70	26	2	0.50																								
32	2.11	89	30	1.88	79	27	1.69	69	24	1.46	59	21	1.25	49	17	1.03	39	12	0.82	30	7	0.63	20	−1	0.42																					
34	2.28	90	32	2.05	81	29	1.85	71	26	1.62	62	23	1.41	52	20	1.19	43	16	0.98	34	11	0.78	25	+5	0.57	16	−3	0.37																		
36	2.46	91	34	2.24	82	31	2.02	73	29	1.79	64	26	1.57	55	23	1.35	46	19	1.13	38	15	0.93	29	10	0.71	21	+3	0.52	13	−6	0.32															
38	2.65	91	36	2.41	83	33	2.20	75	31	1.98	66	28	1.74	58	25	1.54	50	22	1.32	42	18	1.11	33	14	0.87	25	8	0.66	17	+1	0.45															
40	2.85	92	38	2.62	83	35	2.36	75	33	2.14	68	30	1.94	60	28	1.71	52	25	1.48	45	21	1.28	37	18	1.06	29	13	0.83	22	7	0.63	15	−1	0.43												
42	3.06	92	40	2.82	85	38	2.61	77	35	2.36	69	33	2.12	62	30	1.90	55	27	1.69	47	24	1.44	40	21	1.23	33	17	1.01	26	12	0.80	19	+6	0.58	12	−3	0.37									
44	3.29	93	42	3.06	85	40	2.80	78	37	2.57	71	35	2.34	63	32	2.08	56	30	1.85	49	27	1.61	43	24	1.42	36	20	1.19	30	16	0.99	23	11	0.76	16	+4	0.53	10	−5	0.33						
46	3.54	93	44	3.29	86	42	3.04	79	40	2.80	72	37	2.54	65	35	2.30	58	32	2.05	52	29	1.84	45	27	1.59	39	23	1.38	32	20	1.13	26	15	0.92	20	10	0.71	14	+3	0.50	8	−7	0.28			
48	3.80	93	46	3.53	86	44	3.27	79	42	3.00	73	40	2.77	66	37	2.51	60	35	2.28	54	32	2.05	47	29	1.79	41	26	1.56	35	23	1.33	29	19	1.11	23	14	0.88	18	9	0.68	12	+1	0.46	7	−10	0.27
50	4.08	93	48	3.79	87	46	3.55	80	44	3.26	74	42	3.02	67	40	2.73	61	37	2.49	55	34	2.24	49	32	1.98	43	29	1.75	38	26	1.55	32	22	1.30	27	18	1.10	21	13	0.86	16	8	0.65	10	0	0.41
52	4.37	94	50	4.11	87	48	3.80	81	46	3.54	75	44	3.28	69	42	3.02	63	40	2.75	57	37	2.49	51	34	2.23	46	32	2.01	40	29	1.75	35	26	1.53	29	22	1.27	24	18	1.05	19	13	0.83	14	+7	0.61
54	4.69	94	52	4.50	88	50	4.13	82	48	3.84	76	46	3.56	70	44	3.28	64	42	3.00	59	40	2.77	53	37	2.48	48	34	2.25	42	32	1.97	37	29	1.73	32	25	1.49	27	22	1.27	22	18	1.03	17	12	0.80
56	5.02	94	54	4.71	88	53	4.41	82	51	4.11	76	49	3.81	71	47	3.61	65	44	3.26	60	42	3.01	55	40	2.76	50	37	2.51	44	34	2.21	39	32	1.96	34	29	1.71	30	25	1.51	25	22	1.25	20	17	1.00
58	5.37	94	56	5.05	88	55	4.73	83	53	4.46	77	51	4.14	72	49	3.87	66	47	3.55	61	45	3.28	56	42	3.01	51	40	2.74	46	37	2.47	41	35	2.20	37	32	1.99	32	29	1.72	27	25	1.45	23	21	1.24
60	5.75	94	58	5.40	89	57	5.11	83	55	4.77	78	53	4.48	73	51	4.00	68	49	3.91	63	47	3.62	58	45	3.33	53	43	3.04	48	40	2.76	43	38	2.47	39	35	2.25	34	32	1.95	30	29	1.72	26	25	1.50
62	6.14	94	60	5.77	89	59	5.46	84	57	5.16	79	55	4.85	74	53	4.54	69	52	4.24	64	50	3.93	59	47	3.62	54	45	3.32	50	43	3.07	45	40	2.76	41	38	2.52	36	35	2.21	32	32	1.97	28	29	1.72
64	6.56	95	62	6.24	90	61	5.91	84	59	5.51	79	57	5.19	74	56	4.86	70	54	4.59	65	52	4.27	60	50	3.94	56	48	3.68	51	46	3.35	47	43	3.09	43	41	2.82	38	38	2.49	34	35	2.23	30	32	1.97
66	7.01	95	64	6.66	90	63	6.31	85	61	5.96	80	60	5.61	75	58	5.26	71	56	4.98	66	54	4.63	61	52	4.28	57	50	3.99	53	48	3.71	48	46	3.36	44	44	3.08	40	41	2.80	36	38	2.52	32	35	2.24
68	7.48	95	67	7.10	90	65	6.73	85	63	6.36	80	62	5.98	76	60	5.69	71	58	5.31	67	57	5.01	62	55	4.64	58	53	4.34	54	51	4.04	50	49	3.74	46	46	3.44	42	44	3.14	38	42	2.84	34	39	2.54
70	7.98	95	69	7.58	90	67	7.18	86	65	6.68	81	64	6.46	77	62	6.15	72	61	5.75	68	59	5.43	64	57	5.11	59	55	4.71	55	53	4.39	51	51	4.07	48	49	3.83	44	47	3.51	40	44	3.19	36	42	2.87
72	8.51	95	71	8.08	91	69	7.74	86	68	7.31	82	66	6.97	77	64	6.55	73	63	6.21	69	61	5.87	65	59	5.53	61	58	5.19	57	56	4.86	53	54	4.51	49	52	4.17	45	50	3.83	42	47	3.57	38	45	3.23
74	9.07	95	73	8.61	91	71	8.25	86	70	7.80	82	68	7.43	78	67	7.07	74	65	6.71	69	63	6.26	65	62	5.89	61	60	5.53	58	58	5.26	54	56	4.90	50	54	4.54	47	52	4.56	43	50	3.90	39	48	3.54
76	9.66	96	75	9.27	91	73	8.79	87	72	8.40	82	70	7.92	78	69	7.53	74	67	7.14	70	66	6.76	66	64	6.37	62	62	5.99	59	60	5.70	55	59	5.31	51	57	4.92	48	55	4.64	44	53	4.25	41	51	3.96
78	10.28	96	77	9.87	91	75	9.35	87	74	8.94	83	72	8.53	79	71	8.12	75	69	7.71	71	68	7.30	67	66	6.89	63	64	6.47	60	63	6.17	56	61	5.76	53	59	5.45	49	57	5.04	46	55	4.73	43	53	4.42
80	10.93	96	79	10.5	91	77	9.95	87	76	9.51	83	74	9.08	79	73	8.64	75	72	8.20	72	70	7.86	68	68	7.44	64	67	7.00	61	65	6.67	57	63	6.23	54	62	5.90	50	60	5.47	47	58	5.14	44	56	4.81
82	11.63	96	81	11.1	92	79	10.7	88	78	10.2	84	77	9.77	80	75	9.30	76	74	8.84	72	72	8.37	69	71	8.02	65	69	7.56	61	67	7.09	58	66	6.74	55	64	6.39	51	62	5.93	48	60	5.58	45	59	5.23
84	12.36	96	83	11.8	92	81	11.3	88	80	10.8	84	79	10.3	80	77	9.89	76	76	9.39	73	74	9.02	69	73	8.53	66	71	8.16	62	70	7.66	59	68	7.29	56	66	6.92	52	65	6.43	49	63	6.06	46	61	5.68
86	13.13	96	85	12.6	92	83	12.0	88	82	11.5	84	81	11.0	81	79	10.6	77	78	10.1	73	76	9.58	70	75	9.19	66	73	8.67	63	72	8.27	60	70	7.88	57	69	7.48	53	67	6.96	50	65	6.56	47	64	6.17
88	13.94	96	87	13.3	92	85	12.8	88	84	12.2	85	83	11.8	81	81	11.2	77	80	10.7	74	79	10.3	70	77	9.76	67	76	9.34	64	74	8.92	61	73	8.50	57	71	7.94	54	69	7.53	51	68	7.11	48	66	6.69
90	14.79	96	89	14.2	92	87	13.6	89	86	13.1	85	85	12.5	81	83	11.9	78	82	11.5	74	81	10.9	71	79	10.50	68	78	10.0	65	76	9.61	61	75	9.09	58	73	8.58	55	72	8.13	52	70	7.69	49	69	7.25
92	15.69	96	91	15.0	92	89	14.4	89	88	13.9	85	87	13.3	82	86	12.8	78	84	12.2	75	83	11.7	72	81	11.3	68	80	10.6	65	79	10.2	62	77	9.73	59	76	9.26	56	74	8.79	53	73	8.32	50	71	7.84
94	16.63	96	93	15.9	93	92	15.4	89	90	14.8	85	89	14.1	82	88	13.6	79	86	13.1	75	85	12.4	72	84	11.9	69	82	11.4	66	81	10.9	63	79	10.4	60	78	9.98	57	76	9.48	54	75	8.98	51	73	8.48
96	17.63	96	95	16.9	93	94	16.3	89	92	15.6	86	91	15.1	82	90	14.4	79	88	13.9	76	87	13.4	73	86	12.8	69	84	12.1	66	83	11.6	63	82	11.1	61	80	10.7	58	79	10.2	55	77	9.69	52	76	9.17
98	18.67	96	97	17.9	93	96	17.3	89	94	16.6	86	93	16.0	83	92	15.5	79	90	14.7	76	89	14.1	73	88	13.6	70	87	13.0	67	85	12.5	64	84	11.9	61	82	11.3	58	81	10.8	56	79	10.4	53	78	9.90
100	19.77	96	99	18.9	93	98	18.3	89	96	17.5	86	95	17.0	83	94	16.4	80	93	15.8	77	91	15.2	73	90	14.4	70	89	13.8	68	87	13.4	65	86	12.8	62	85	12.2	59	83	11.6	56	82	11.0	54	80	10.6

Values of temperature and humidity within the usual limits required for ventilation are given above. Fuller information can be obtained from the Psychrometric Chart Table VI., and from more extensive tables in "Fan Engineering" (Carrier), "Heating and Ventilation" (Sturtevant), etc., etc.